HYBRID AND CLOSED CYCLE PNEUMATIC POWER PLANTS

CONTENTS

THE ARNOLD ELECTRO-PNEUMATIC RAILWAY SYSTEM: ITS APPLICATION AND EXPERIMENTS THEREWITH IN CONNECTION WITH THE LANSING, ST. JOHNS & ST. LOUIS RAILWAY.

By BION J. ARNOLD.

[IN a preliminary statement Mr. Arnold alludes to his persistent advocacy of the use of the alternating current directly in motors for electric railways, for several years past, and to the fact that but few engineers in this country advised it. Only recently has the three-phase system been advocated and demonstrated abroad, but now important advances have been made in the development of a single-phase system,

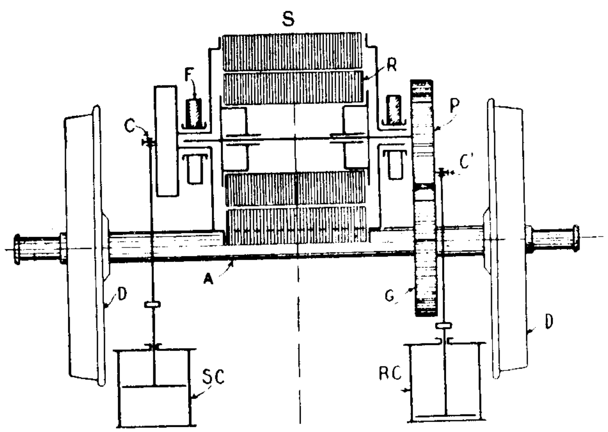

FIG. 1.—DIAGRAM OF TRUCK.

which is expected to work a revolution in electric railway work.

Since the announcement of the principles of his single-phase system before the Great Barrington convention Mr. Arnold has been conducting experiments on a large scale at his own private expense, expecting to celebrate the incoming of the year 1904 with a public demonstration over a road twenty miles long, with the idea of showing that a single-phase electric railway is not only operative, but efficient, and less costly than any other system of the direct-current type. But his expectations were met by disappointment, in consequence of a destructive fire occurring on December 18, 1903, at Lansing, Mich., in the car barns, which destroyed two new cars built for his system, and so damaged an experimental electric locomotive as to render it inoperative, thereby necessitating an abandonment of the proposed test.

Mr. Arnold is convinced from experiments so far conducted that the year 1904 will be an epoch-making one, marking a revolution from the direct-current to the alternating current for railway work. He predicts there will be a beginning on a large scale of the displacement of the steam locomotive on railways by the use of a substantial form of overhead construction, rather than the third rail. He intends to give the results of his experiments at a later date before the American Institute of Electrical Engineers. In January, 1900, he examined a route for a road between Lansing and St. Louis, Mich., a distance of sixty miles. By November 15, 1901, twenty miles of road were completed to St. Johns, Mich., over which steam trains were operated. After much delay the electrical equipment was completed for this section by December 15, 1902.

On June 15, 1903, two trips were made, each three miles long, with his first experimental machine. On the first trip seven persons were carried, and thirteen on the second.

The correctness of the theory of the operativeness of the improvement was demonstrated; but owing to the somewhat crude electro-pneumatic motor, full and efficient tests could not be obtained. Since these trials a double-equipped truck has been perfected, which was built in the form of a locomotive, and this was entirely completed with test instruments, ready for operation, when the fire occurred. The main economy claimed consists in keeping a constant, uniform load on the motor. Mr. Arnold now describes fully his system as explained below. We are indebted to him for plans, photographs, and details.—ED.]

ROADBED AND TRACK.

The Lansing, St. Johns & St. Louis Railway was originally projected to extend from Lansing, the capital of Michigan, northward through St. Johns, Alma, and St. Louis, a distance of about sixty miles, but up to the present time only that portion extending from Lansing to St. Johns, a distance of twenty miles, has been constructed.

This road was built in accordance with steam railroad practice, with easy grades and curves, so that steam locomotives could be operated over it until such time as electrical equipment could be put upon it; the idea being to complete the road in such a manner that it could be utilized for both freight and passenger service, and thus secure all the business available from the territory through which it passes.

The road is equipped with 67-pound T-rails, laid on ties spaced 2 feet apart between centers, and as alternating high-tension current was to be used, but one of these rails was bonded with 38-inch 4/0 bonds extending entirely around the splice bars.

Since it was impossible to secure rails from the rail manufacturers in time, rails and splice bars were secured from one of the leading steam railways, and this necessitated the adoption of a supported joint and a long bond, as there was not room under the splice bars for concealed bonds.

The road as at present constructed between Lansing and St. Johns has no grades exceeding 1 per cent, and no curves exceeding 7 degrees, except in the cities themselves, where the terminals of the road run over

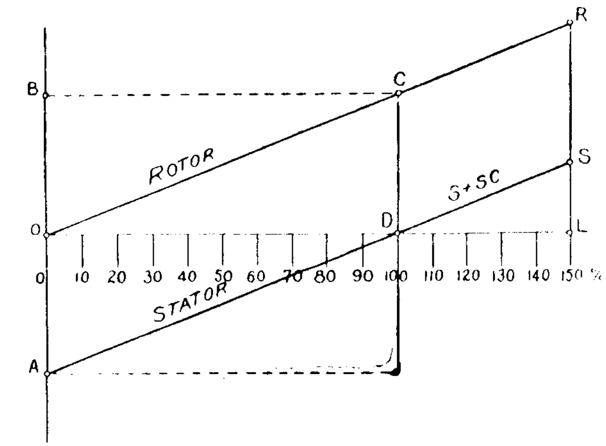

FIG. 2.—DIAGRAM OF OPERATION.

the streets and make such curves as ordinary street cars make, the minimum radius being 50 feet. At each city a terminal was planned, so that all freight would be diverted to connecting steam roads, thus making it unnecessary for the freight service to pass over the city streets or curves.

At the Lansing end it was necessary to pass over the steam railway tracks of the Pere Marquette Railroad, and this necessitated the construction of a bridge, with pile approaches. The grade as approached from the Lansing end was 4 per cent for a distance of about seven hundred feet, and after passing over the bridge the descending grade is 2.3 per cent for about five hundred feet. At the St. Johns end there is a grade on the principal street of the town averaging about 2 per cent for about fifteen hundred feet.

OVERHEAD CONSTRUCTION.

Considerable care was taken in planning a suitable insulator for carrying the trolley wire, an annealed glass insulator being used.

In the overhead construction wood is used for the pole, crossarm, and brace, and the insulator is supported by means of a short span wire from iron brackets secured to the wooden crossarm. This construction insures a high insulation at a low first cost, the entire line having been constructed at but a slightly increased expense over the cost of standard construction, and at the same time so built that in case of failure of the alternating motor system the standard direct-current motor system could be put into service without changing any parts. Even holes for the pins for carrying the extra feeders which would be required were provided.

The line and track work were constructed in such a manner that no expense was incurred for any parts which would not be required for standard construction in case it became necessary to ultimately adopt the standard direct-current motor system; the entire idea

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in the construction of the road being to save first cost and to invest all that was invested in such a manner that all material purchased would be utilized in case either system were adopted; and should the alternating system prove successful, the additional investment for a direct-current motor system need not then be installed.

The working conductor was placed twenty-two feet above the top of the rails, in order that trainmen when standing upon the tops of the freight cars going over the road could not come in contact with the working conductor.

It was planned to operate the entire road from a single 00 trolley wire, and with one rail bonded as hereinbefore mentioned; this amount of copper being sufficient to operate four 40-ton cars at an average speed of thirty miles per hour with power house located one and one-half miles from one end of the line, and operating with from six thousand to ten thousand volts on the working conductor.

The power house is located at one end of the line, owing to the electric company from which power is purchased by the railroad having a water power at this point. Current is transmitted to the nearest end of the line over two No. 3 wires. The power is furnished from a 300-kilowatt rotary converter generating at 380 volts, at 25 cycles, the energy from which is stepped up to the working pressure of the line. It was the intention, after experimenting a sufficient length of time to determine the best voltage for the working conductor, to have the generators for the permanent plant constructed so as to generate at this determined voltage, and it was for this reason that a temporary rotary converter was first installed to conduct the experiments with.

During the preliminary experimental period upon the apparatus hereinafter described, all power was transmitted from the above-mentioned power house to a point about two miles distant, where were located the car barns in which the preliminary experiments were made.

The conditions under which the first application of the system took place having thus been set forth, it may be well, in order to get clearly before the reader the principles on which the system is based, to quote here the statements made by Mr. Arnold before the Great Barrington convention on June 19, 1902, as follows.

"The principles underlying the system I advocate, and which I call an electro-pneumatic system, are as follows:

"1. A single-phase or multiphase motor, mounted directly upon the car, designed for the average power required by the car, and running constantly at a constant speed and a constant load, and, therefore, at maximum efficiency.

"2. Instead of stopping and starting this motor and dissipating the energy through resistances, as is customary with all other systems known to me, I control the speed of the car by retarding or accelerating the parts usually known as the rotor and stator of the motor, by means of compressed air, in such a manner that I save a portion of the energy which is ordinarily dissipated through resistances, and store it to assist in starting the car, helping over grades, for use in switching purposes, and for the operation of the brakes.

"3. By this method of control I secure an infinite number of speeds from zero to the maximum speed of the car, which may or may not be at the synchronous speed of the motor, for with the air-controlling mechanism working compressing, the speeds below synchronism are maintained, and by reversing the direction of the air through the controller speeds above synchronism may be attained for reasonable distances. This feature gives to the alternating-current motor the element absolutely essential for practical railway work, for it permits a car or train to ascend a grade at any speed with the motor working at its maximum efficiency and imparting its full torque to the car. When descending the grade the motor may utilize its full power drawn from the line in compressing air, or it may be used to compress air with the stored energy of the train, thereby acting as a brake.

"4. By virtue of the air-storage feature, each car becomes an independent unit and capable, in case of loss of current from the line, of running a reasonable distance without contact with the working conductor. This feature will enable a car to work on a high-tension trolley wire or active conductor over private right of way, and allow the active conductor to be stopped where the private right of way ceases, and the car to proceed through a city or town on any tracks, whether electrically equipped or not, until it reaches the outskirts of the city or town, where it can take up the working conductor again on private right of way. This feature is also valuable in switching work, for each car being independent, it can leave the main-line track and operate over switches or sidings without complicating the yards with additional overhead or third-rail conductors, thus necessitating through-line conductors over main-line track or tracks only.

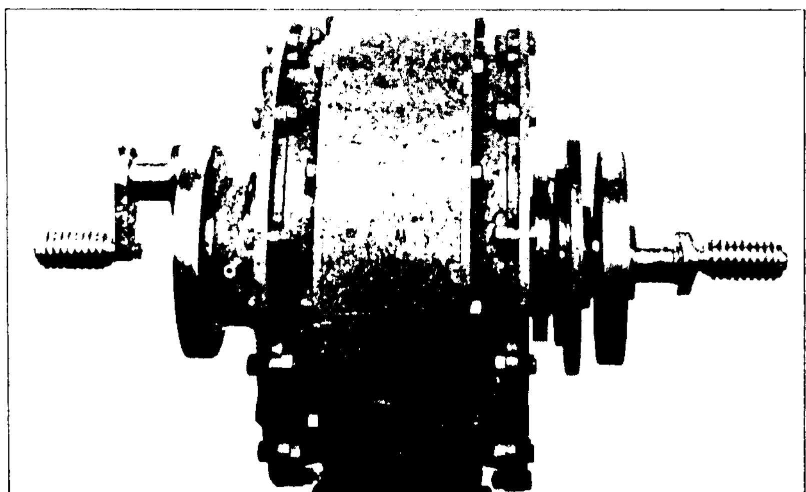

FIG. 3.--OUTSIDE VIEW OF ELECTRIC MOTOR.

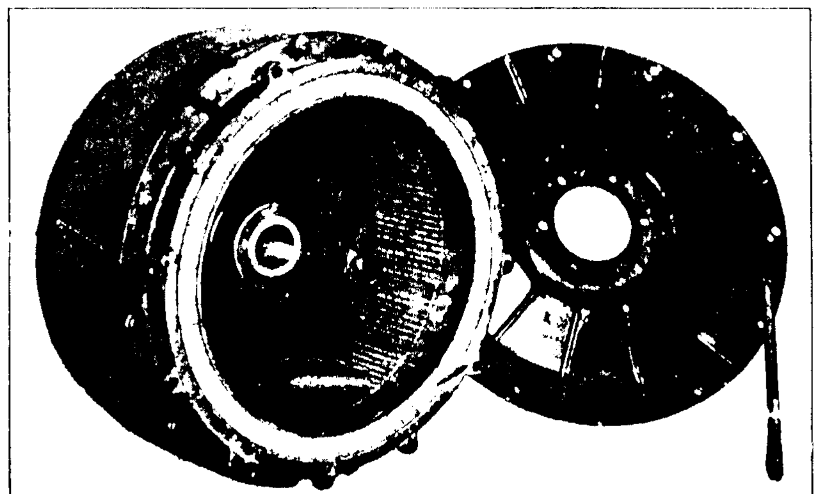

FIG. 4.--INTERIOR OF ELECTRIC MOTOR.

"5. Since a single-phase motor can be used, the motors can be supplied with current from a single overhead wire or third rail, and with a single-rail return circuit, thus permitting the overhead construction, or third-rail construction, to conform to the standard to to-day, except that a much higher working voltage can be used, provided the insulation is taken care of. Furthermore, in steam railway work this system, by virtue of its single-phase feature, will only require the use of one of the track rails for the return circuit, thus leaving the other rail for the use of the signal system, which up to the present time, does not seem to have been satisfactorily solved without the use of one of the track rails.

"6. The current will be take.. from the working conductor at any voltage up to the limit of the insulation, and in case this voltage is high (I am building my line for 15,000 volts), a static transformer will be carried upon each car, and the pressure reduced from the line voltage to the voltage of the motor, which in the case under consideration is designed for 200 volts. Where it is unnecessary to utilize so high a line pressure, the motor may be designed for the working voltage, and the current fed directly from the working conductor into the motor, thus eliminating the static transformer. When a high--voltage working conductor and static transformer are used, and it is thought advisable to use a working conductor through cities or towns, this working conductor will be supplied with energy through a stationary transformer at each city limit, thus making the working conductor through the cities or towns safe.

"7. By virtue of the speed of the motor and its constant load, either when the car is in motion or when it is standing still, and the motor is compressing air, the variable load now customary in electric-railway power plants is eliminated, and the power station works at practically a constant load, thereby eliminating a large part of the investment at present requisite in power station and line construction. Furthermore, by virtue of the air-storage feature, each car, in the particular apparatus I have designed, is capable at any time, when current is on the working conductor, of delivering to the car wheels a much greater torque in proportion to the capacity of the motor than is possible with any electrical system known to-day.

"I believe that by the adoption of this system the following results will be accomplished:

"1. The entire elimination of the present standard system of rotary converter sub-station plant, together with the maintenance thereon, and the cost of the necessary attendants.

"2. The absorbing and rendering available for useful work in starting, or otherwise, of a large percentage of the energy stored in the moving mass, which, under the present methods of operation is dissipated at the brake shoes.

"3. A large reduction in the first cost of electrically equipping long-distance railroads, thereby making it feasible, from an engineering and business standpoint, to equip many roads which cannot now be shown advisable, thus opening up the steam railway field to the industry in which we are now engaged."

The following description will explain more in detail the application of the principles of the system and the mechanism of its working parts:

Fig. 1 represents diagrammatically the working parts of one form of the system. The rotor R of a single-phase induction motor is geared to the axle of the car, and by means of crank pin C' secured in pinion P also drives the compressor cylinder $R\ C$, while stator S can freely revolve around the rotor, and drive by means of crank pin C the compressor cylinder $S\ C$. Both cylinders are piped to air reservoirs located under the car, and are also provided with suitable valves manipulated from a single controller on the car platform for making them perform their various functions; thus the entire regulation of the speed and power of the car are controlled by the air cylinders, and no other regulating devices are necessary. The cylinder valves are electrically operated, which makes it possible for each cylinder when driven by the electric motor to compress air into the tanks, and when operated by compressed air to furnish mechanical energy for moving the car. When, for instance, the cylinder is compressing air, the valves work like inlet and outlet poppet valves of a common air pump, while on the other hand, if the cylinders are supplied with compressed air, each valve is operated electrically by a pilot solenoid connected with the valve seat in such a manner that the energy for moving the valve is supplied by the compressed air, thereby making the valve practically self-actuating. The time of operation of the valves is controlled by a series of collector rings revolving with the engine shaft, and their regular operation is interrupted and varied to suit the requirements

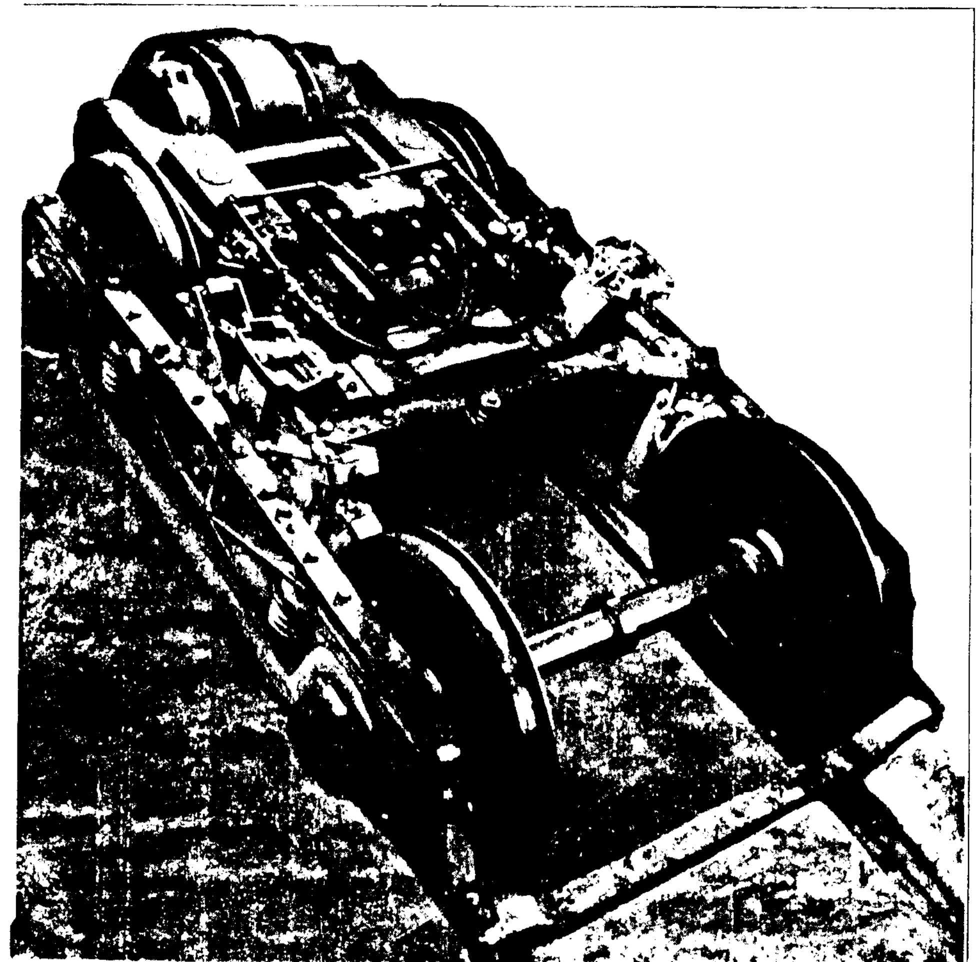

FIG. 5.—FIRST EXPERIMENTAL MOTOR WITH MOTOR FORWARD.

by means of the motorman's controller.

When a rotary or turbine type of air engine is used, all of the above valves and reciprocating parts

FIG. 7.—COMPLETE CAR.

are eliminated, and their entire controlling mechanism consists of two air valves operated from a single engineer's valve, which may be located upon the platform of the car or in the cab of the locomotive, and so arranged that one or more units may be operated from the platform or cab of any unit without the necessity of connecting wires between the units.

Since the motor may be of the simplest types of induction motor without a commutator, and the system does not require the manipulation or breaking of the main current, the motor may be designed for any working voltage and be of any type which will maintain a constant speed when provided with a constant load. This eliminates the necessity of all step-down transformers, resistances, or other regulating devices, and confines the current to the motors themselves, and as these are below the car floor, the danger from the current is reduced to the minimum.

At the same time, the air cylinders, in addition to performing all the functions of speed control, give to the machine the independent unit element, and the ability to store the kinetic energy of the train in stopping and utilize it in starting. On account of these and other features, the electric motors of this system can be much smaller in capacity, when rated as continuous working motors, than those of other systems not possessing this equalizing load feature, and the capacity of the power house and line can be reduced to about one-half of what would be required with systems where the fluctuating starting loads of the cars are transmitted back to the power house.

In order to better understand the different operations of the system, Fig. 2, showing a speed diagram, has been prepared, in which on the axis of abscisses $O\ D\ L$ are represented the different car speeds in per cent. of the synchronous motor speed, while the co-ordinate axis $A\ O\ B$ represents the rotor and stator speeds corresponding to the car speeds shown.

The operation of the car may be divided into the following periods:

1. *Standing in the Station.*

Referring to Fig. 1, the rotor R is standing still, while the stator S runs with full synchronous speed. The stator is then transferring the full energy of the electric motor through crank C to the compressor cylinder $S\ C$, which energy is being delivered in form of compressed air into the air reservoir.

Since the relative velocity between the stator and the rotor is, under all conditions of operation, constant, the speed curves of stator and rotor may be represented by two parallel lines $O\ C\ R$ and $A\ D\ S$ in Fig. 2. The origin O of the given co-ordinate system represents the period of rest of the car, and therefore indicates zero rotor speed and full stator speed in a negative or downward direction, as the stator is now revolving in the opposite direction from that which the rotor must revolve to drive the car forward.

Let it be further assumed that for an instant $O\ A$ equals the active torque of the stator, then it will be easily understood that $O\ B$, which equals $O\ A$, represents the reactive torque of the rotor exerted on the car axle, meaning that if the car is free to move, the reactive torque can be used advantageously for the starting and acceleration of the car.

When the car is standing in a station, it is held at rest by moving the controller to such a position that the outlet pipe from rotor cylinder $R\ C$ is throttled, thereby increasing the pressure behind the piston to such an extent that it overcomes the effort of the rotor R to revolve, thus tending to cause the stator S to revolve, and at the same time holds the car at rest without the use of wheel brakes.

2. *Starting and Acceleration.*

To start the car, the air cushion behind the piston of $R\ C$ is removed, and the air which is being compressed by cylinder $S\ C$, supplemented by the stored air from the tanks, is admitted to cylinder $R\ C$ with the controller at the position of maximum cut-off. The rotor then begins to revolve, and as it accelerates, the stator slows down by exactly the same amount that the rotor has increased its speed; and as the rotor and car speed increases, the controller is gradually moved to a smaller percentage of cut-off until the car speed corresponds to the full synchronous speed of the motor,

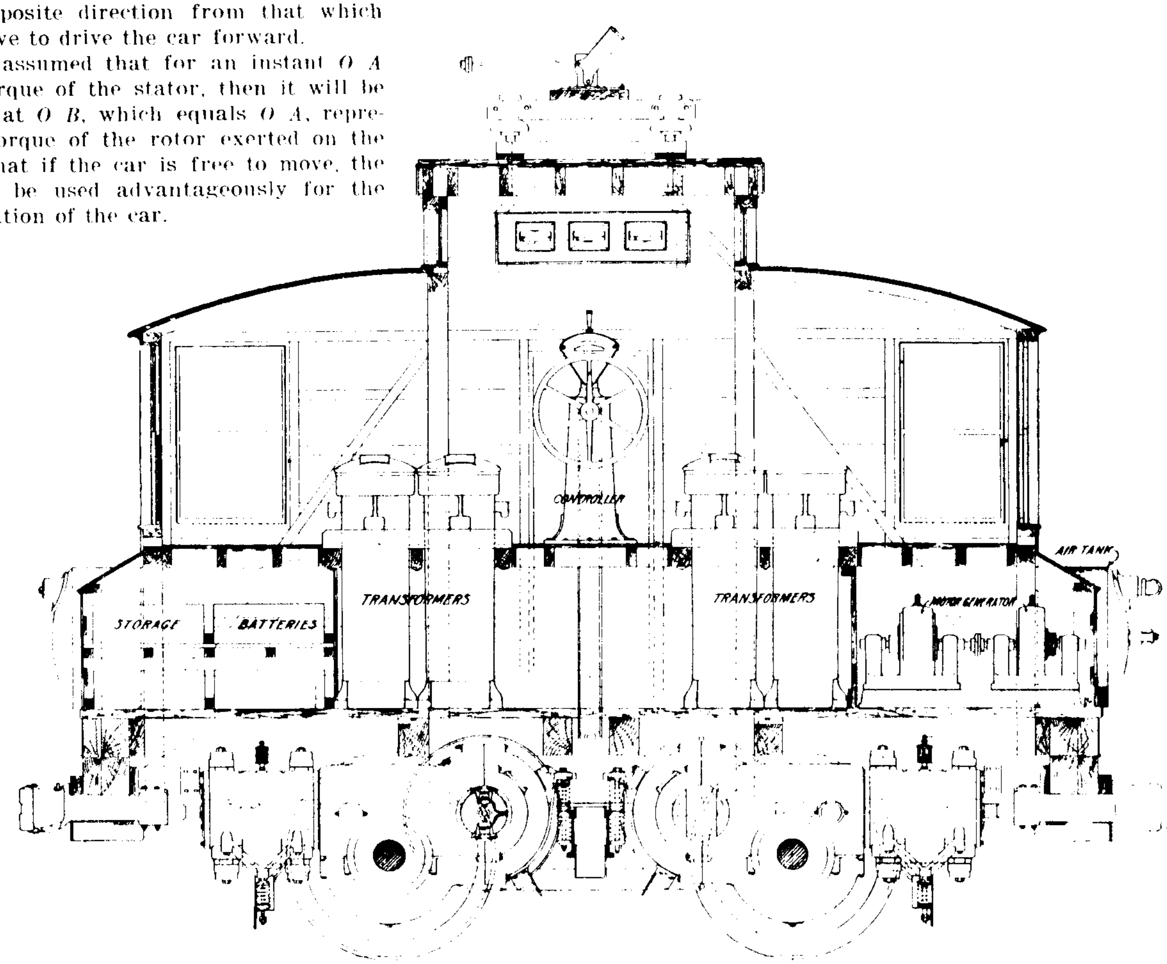

FIG. 6.—LONGITUDINAL SECTION OF ELECTRIC LOCOMOTIVE.

at which time the stator comes to rest.

During this period of acceleration the air compressed by cylinder *S C*, instead of being delivered to the tanks to lose its heat, is delivered, hot, directly to the rotor cylinders, thus greatly increasing the efficiency of the combination, as the heat usually lost in air systems is utilized and the advantages of heated air gained without a reheater; and as the pressure used is low, many of the ordinary difficulties in the use of compressed air disappear. If the rate of acceleration is such that cylinder *R C* uses all of the air supplied by the cylinder *S C*, no exhaust to the atmosphere from cylinder *R C* takes place.

Referring now to Fig. 2, which graphically represents this process, since the electric motor runs always at a constant speed and a constant load it has a constant torque, and therefore the distance between lines *O C R* and *A D S* may be considered as representing the energy delivered by the electric motor.

The length of any ordinate extending from *O D* to *O C* represents the proportionate amount of energy derived from the electric motor which is applied directly through pinion *P* and gear *G* of Fig. 1 to the propulsion of the car, while the corresponding ordinate extending below *O D* to *A D* represents the proportionate amount of the energy of the electric motor which is absorbed in compressing air through cylinder *S C*, which energy, in the form of air, is immediately transferred to cylinder *R C* and is utilized in accelerating the car.

In practice, however, since there will be a loss in transferring the energy from electrical energy to energy in the form of compressed air and back again into mechanical energy, this loss, whatever it may be, must be drawn from the storage tanks, and the requisite amount of air from these tanks supplied to rotor cylinder *R C*, in order to maintain the full power of the electric motor upon the car axle during the period of acceleration. Should it be desired to accelerate at a greater rate than the full power of the electric motor is capable of giving to the car, the additional energy may be supplied in the form of air from the storage tanks through cylinder *R C*, thus increasing the total energy given to the car during acceleration, in which case this total power would be represented for any given instant by a point above line *B C*.

3. *Full Speed.*

When the rotor has reached full synchronous speed, by the previous operation, this speed can be maintained by moving the controller to another position, which will throttle the outlet pipe of cylinder *S C* until the reaction due to the pressure behind the piston equals the full capacity of the electric motor. An overload or underload may be placed upon the motor by varying this pressure, but under normal conditions of operation cylinder *S C* is provided with an automatic valve which keeps a constant pressure behind its piston, thus maintaining an absolutely constant load upon the electric motor, and consequently a uniform demand of electrical energy from the line. This uniform load is represented by the parallel lines *O C R* and *A D S* of fig. 2.

With the controller set at full-speed position, the inlet valves of rotor cylinder *R C* are held open, and the piston runs free, and the electric motor now gives its full power to the car axle, and the stator and its air mechanism will remain at rest as long as the car runs at the speed corresponding to the synchronous speed of the motor.

4. *Speed Variations.*

There are usually certain places on any road where high rates of speed can be maintained for short distances, and as these speeds might be higher than the synchronous speed for which the motor was designed, they are provided for as follows:

Assuming that the car is running at synchronous speed, the controller may be moved to such a position that the valves of stator cylinder *S C* operate in such a manner as to cause it to act as an engine and revolve stator *S* in the same direction as rotor *R* is revolving. This now causes, owing to the constantly electrically-maintained relative difference in speed between the stator and the rotor, an increase of speed of the rotor and car axle, due to the motor automatically working as a magnetic clutch, without mechanical contact; and if the resistance of the car or train is less than the capacity of the electric motor, the air necessary for revolving the stator can be obtained, hot, from the rotor cylinder *R C* without drawing from the tanks, and a speed above synchronism indirectly proportioned to the resistance of the train, maintained indefinitely. When the resistance of the train is greater than the capacity of the electric motor, speeds above synchronism can be obtained only by supplying rotor cylinder *R C* with stored air from the tanks, and can only be maintained for short distances, or until the storage capacity of the air reservoirs is exhausted. This condition corresponds to the spurts that can be made by a steam locomotive when working above the steaming capacity of the boiler. The distance from the line *O D L* to that portion of the line *A D S* above *O D L* in Fig. 2 represents, at any given speed, the proportionate amount of energy which must come from the tanks and be supplied through cylinder *S C*; and the distance from *D L* to *C R* represents the total energy given to the car by the combined action of the electric motor and the stator cylinder when operating under these conditions.

The energy delivered to the car can be still further increased by admitting air into rotor cylinder *R C* and allowing it to work as an engine.

5. *Retardation.*

To bring the car or train to rest, instead of applying mechanical brakes to the wheels in the ordinary manner, and thereby dissipating the entire stored energy of the car or train in the form of heat, this energy is saved in the form of compressed air, to assist in starting the car or train, by setting the controller in such a position that rotor cylinder *R C* compresses air and delivers it into the storage tanks. Any desired rate of retardation can be secured by throttling the delivery pipes from rotor cylinder *R C*, and in practice this pipe is provided with an automatic valve which releases just before the slipping point of the wheels, thus allowing the motorman to brake as rapidly as he desires without liability of flattening the wheels. Supplemental wheel brakes are provided for emergency, but need not often be used, and the ordinary wear and tear on them saved. When the car is again at rest, the cycle of performance as above given is repeated for the next run.

6. *Reversing.*

When it is desired to run the car backward for short distances, the electric motor is not disturbed, and the power is furnished by the rotor cylinder *R C* by reversing the action of the valves; but if it is desired to run backward for any great distance, the current is thrown off the motor, the stator engine reversed, and the stator brought to speed by the air, when the current is again thrown on to the motor, and the cycle of operation is the same as when running forward.

Fig. 3 represents the exterior of the electric motor, showing the cranks of the stator and rotor, also collector rings for operating the valves of the air cylinders when working as engines.

Fig. 4 shows an interior view of the stator of the motor with the flange removed, the rotor of the motor being of the standard squirrel-cage induction type.

Fig. 5 shows, mounted upon a truck, a view of the first electro pneumatic motor constructed, and upon which the first experiments were conducted.

Since the single motor represented in Fig. 5 was too small in capacity to propel so large a car, it was decided to experiment with an improvised locomotive, consisting of the truck and motors similar to Fig. 6, carrying suitable air tanks and transformers upon a temporary frame structure. With an equipment of this kind, the trial runs were made and passengers carried on June 15, 1902.

Fig. 6 shows a double truck fitted up in the form of a locomotive, in longitudinal section; it was this locomotive that was recently destroyed by fire. In order that the locomotive might operate as an independent air unit upon tracks not equipped with overhead electrical conductor, it was provided with a small storage battery and small generator for charging the batteries and for operating the headlight. These auxiliaries are not necessary for the successful operation of the system, provided the locomotive can always be supplied with electric current from the working conductor, for then the valves can be made to operate from alternating current and thus eliminate the use of motor-generator

and batteries. When, however, it is desired to operate independently of the electric conductor, these auxiliaries are necessary, and one set may supply an entire train. It will be seen that the locomotive is also provided with transformers, another auxiliary which is unnecessary in case the motors are designed for the voltage transmitted over the working conductor; but in this case transformers were used because the manufacturer of the motors could not be induced at the time they were purchased to build a high-tension motor for railway work consequently the parts of a standard motor were utilized, and a pressure of 200 volts adopted for the motors, as this was the most economical voltage that could be used with the particular parts selected. This locomotive was provided with all necessary testing instruments, and had been operated in the barns for some time and found to perform all its functions successfully, and would have been placed on the road, and experiments with it would now be in process had it not been destroyed.

ALEXANDER FUELESS CAR

In a matter of 45 days Robert Alexander and his partner with a budget of $500.00 developed a car that virtually confounded the experts. A very small engine provided the power. The engine was actually recharged once it was moving, by a hydraulic and air system. This actually took care of the small electric drain.

Determined that the auto industry would not burry their invention, according to the Monelliello, CA. inventors last reports, but what happened to U.S. PAT. #3913004???

United States Patent [19]

Alexander

[11] **3,913,004**

[45] **Oct. 14, 1975**

[54] **METHOD AND APPARATUS FOR INCREASING ELECTRICAL POWER**

[75] Inventor: **Robert W. Alexander**, Pasadena, Calif.

[73] Assignee: **Alex**, Pasadena, Calif.

[22] Filed: **Nov. 18, 1974**

[21] Appl. No.: **524,556**

[52] **U.S. Cl.** **321/28**; 321/50

[51] **Int. Cl.²** **H02M 7/64**

[58] **Field of Search** 310/113, 165; 321/28, 29, 321/30, 31, 48, 49, 50

[56] **References Cited**

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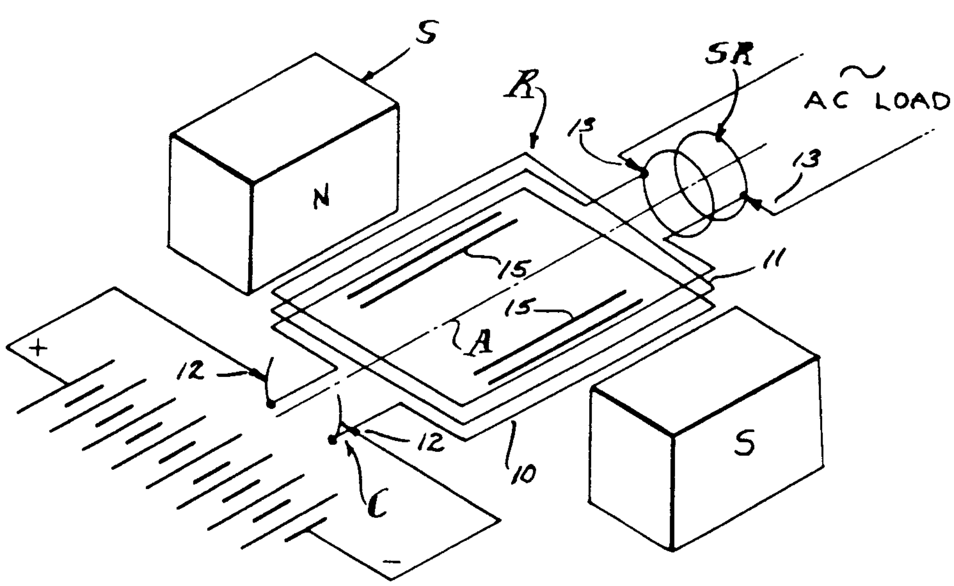
Primary Examiner—William M. Shoop

[57] **ABSTRACT**

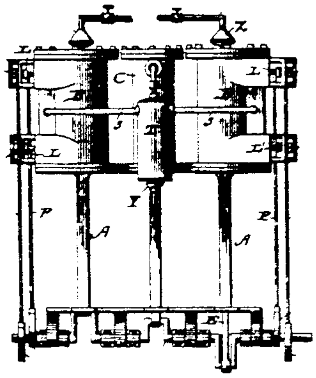
A form of rotating machine arranged in such a way as to convert a substantially constant input voltage into a substantially constant output voltage; involving generally a rotor that revolves at substantially constant speed within a stator and which comprises a transformer core subjected to and having a primary motor-transformer winding and a secondary transformer-generator winding; whereby transformed and generated power are synchronously combined as increased output power.

27 Claims, 3 Drawing Figures



JANUARY 5, 1904.

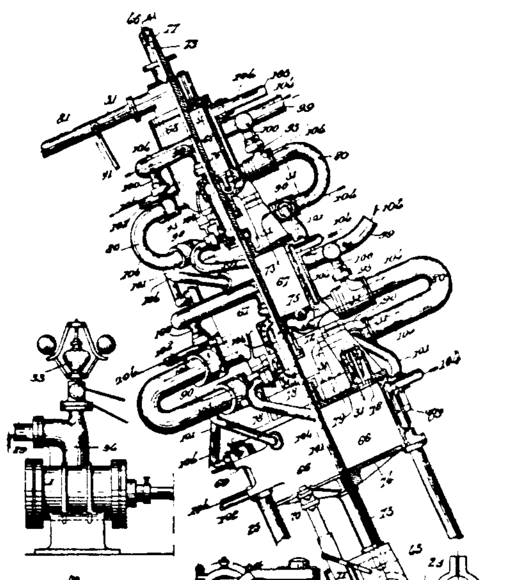
748,660. **COMPRESSED-AIR MOTOR. EDWARD W. SCHLOEMER, Milwaukee, Wis. Filed Mar. 30, 1901. Serial No. 53,662. (No model.)**



Claim.—A motor comprising two motor-cylinders, pistons therein, each motor-cylinder having two parallel longitudinal passage-ways opening into the cylinder on each side of the piston, rotary valves in said passage-ways adjacent each end thereof, a compressor-cylinder of less diameter than the motor-cylinders and arranged between them, a reservoir-tank, a piston in the compressor-cylinder, pipes leading from each end of the compressor-cylinder to the tank, and thence discharging into one of the longitudinal passages of each motor-cylinder, an exhaust-pipe leading from the other passage, a cranked shaft, piston-rods connected at one end to the pistons, and at their opposite ends to the shaft, two eccentrics arranged on the shaft adjacent each motor-cylinder, eccentric-rods extending parallel with each motor-cylinder, oppositely-arranged valve-stems, connected at one end to the valves in one of the passages and at their opposite ends to one of said valve-rods, means for initially compressing the air in the tank and means for heating the motor-cylinders.

AUGUST 14, 1906.

828,295. COMPRESSED-AIR MOTOR. IVAN W. AMMON, St. Petersburg, Russia. Filed Sept. 24, 1903. Serial No. 174,411.

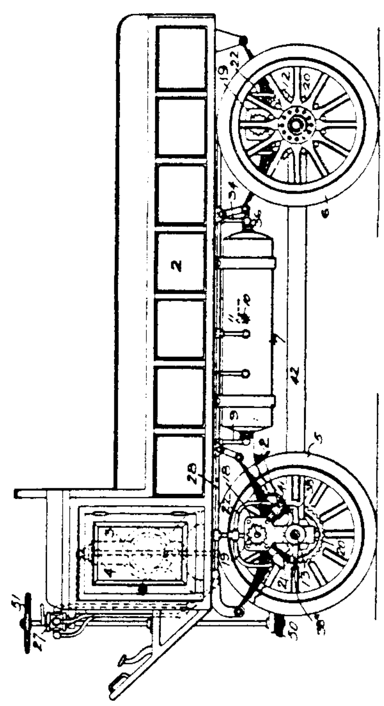


Claim.—1. The combination in a motor operated by compressed and heated air of three cylinders in which the compressed and heated air expands successively, of a crank-shaft, driven through connecting-rods by the piston-rods of the driving-pistons, two air-compressors, operated by means of connecting-rods by the said main shaft, the axial line of each of these compressors making a suitable angle with the working axial line of the cross-head operating the pistons of the corresponding compressor by means of a connecting-rod, the length of this connecting-rod, the stroke of the compressor-piston and that of the cross-head being equal, as described and shown.

2. The combination in a compressed and heated air motor with three motor-cylinders, of a crank-shaft and two air-compressors, of pipes leading the compressed air to an air-heating apparatus, a pipe leading the heated and compressed air to the distributer of the smallest of the three motor-cylinders; a pipe leading the expanded air from the largest of the three motor-cylinders to a chamber and a pipe leading air from this chamber to the aforesaid compressors, as described and as shown.

JULY 9, 1907

859,235. AUTOMOBILE. WALTER W. MACFARREN, Pittsburg, Pa., assignor to William H. Donner, Pittsburg, Pa. Filed June 18, 1906. Serial No. 322,207.



1. In a motor vehicle, the combination of a prime motor, an air compressor, the motor being adapted to drive the compressor, an air motor adapted to propel the vehicle, means for supplying the compressed air to said motor, and connections between the exhaust port of the air motor and the intake port of the compressor.

2. In a motor vehicle, the combination of a prime motor, an air compressor adapted to be driven by said motor, a reservoir adapted to receive the compressed air from said compressor, an air motor adapted to receive its air from said reservoir and to drive the vehicle, and means for connecting the exhaust port of the air motor with the intake port of the compressor.

3. In a motor vehicle, the combination of a prime motor, an air compressor adapted to be driven by said motor, a reservoir adapted to receive its compressed air from said compressor, an air motor adapted to receive its air from said reservoir and to drive the vehicle, a closed passage leading from the exhaust port of the air motor to the intake port of the compressor, and a non-return valve opening from the outer air to said passage.

4. In a motor vehicle, the combination of a prime motor, an air compressor adapted to be driven by said motor, a reservoir adapted to receive its compressed air from said compressor, an air motor adapted to receive its air from said reservoir and to drive the vehicle, a closed passage leading from the exhaust port of the air motor to the intake port of the compressor and a valve adapted to open said passage to the outer air at a predetermined pressure in said passage.

5. In a motor vehicle, the combination of a prime motor, an air compressor adapted to be driven by said motor, an air reservoir adapted to receive its air from said compressor, and an independent motor on each of a plurality of wheels of said vehicle adapted to drive the same, the air motors being adapted to receive their air from said reservoir.

6. In a motor vehicle having four driving wheels, the combination of a prime motor on said vehicle, an independent secondary motor for each of said driving wheels mounted on the axles adapted to drive said wheels, and means for conveying the power from said primary motor to said secondary motor.

7. In a motor vehicle, the combination of a prime motor, an air compressor adapted to be driven by said prime motor, an air reservoir on said vehicle, an air motor adapted to propel the vehicle and to receive its air from said reservoir, and means for connecting said reservoir with the cylinder of said gas engine to start said engine.

8. In a motor vehicle, the combination of a prime motor, an air compressor adapted to be driven by said prime motor, triangular shaped double cylinder motors on the axles, connections between said motors and the compressor, and connections between said motors and the wheels.

9. In a motor vehicle, the combination of a prime motor, an air compressor adapted to be driven by said motor, an air reservoir on said vehicle, an independent air motor on each of the wheels of said vehicle adapted to receive their air from said reservoir, a controlling valve suited to control all said motors, and suitable pipes connecting said reservoir and air motors.

10. In a motor vehicle, the combination of a pivoted wheel, a motor mounted on the axle of said wheel so as to turn therewith about its pivot, a prime motor on the vehicle, and means for conveying the power from said prime motor to the motor upon the axle.

11. In a motor vehicle provided with two or more wheels having their axles pivoted to said vehicle, a driving motor upon each of said axles, a prime motor upon the vehicle, and means for conveying the power of said prime motor to the motors upon said axle.

12. In a motor vehicle having two or more driving wheels, the combination of a prime motor on said vehicle an air compressor on said vehicle adapted to be actuated by said prime motor, and an independent secondary air motor upon the axles for each of said driving wheels adapted to drive the same, and means for conveying the compressed air from said compressor to said secondary motor.

13. In a motor vehicle, the combination of a pivoted wheel, a compressed air motor mounted on the axle of said wheel so as to turn therewith about its pivot, a prime motor on the vehicle, an air compressor on said vehicle adapted to be operated by said prime motor, and means for conveying the compressed air from said prime motor to the motor upon the axle.

14. In a motor vehicle provided with four wheels having their axles pivoted to said vehicle, a compressed air driving motor upon each of said axles, a prime motor upon the vehicle, an air compressor upon said vehicle adapted to be operated by said prime motor, and means for conveying the compressed air from said compressor to the motors upon said axle.

15. In a motor vehicle, the combination of a pivoted wheel, a compressed air motor mounted on the axle of said wheel so as to turn therewith about its pivot, a prime motor upon the vehicle, an air compressor on said vehicle adapted to be operated by said prime motor, a flexible conveying pipe communicating with the intake of the compressed air motor and with the discharge port of said compressor.

16. In a motor vehicle, a prime motor, an air compressor driven by said prime motor, one or more air motors adapted to drive the wheels, and a subdivided reservoir intermediate said air motors and said compressor so arranged that a small portion of the same only may be in connection with the air motors and compressor.

17. In a motor vehicle, a prime motor, an air compressor driven by said prime motor, one or more air motors adapted to drive the wheels, a subdivided reservoir intermediate said air motors and said compressor, and a valve controlling the connection between the subdivisions of said reservoir.

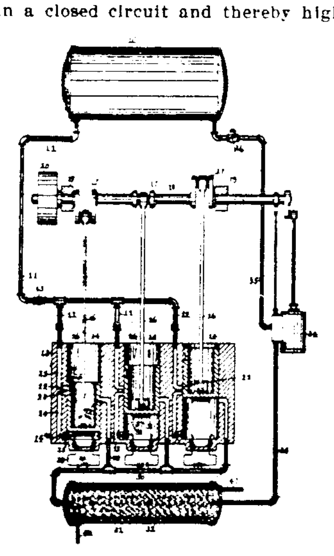
18. In a motor vehicle, the combination of an axle, a stub-shaft, a wheel mounted on said stub-shaft, a motor supported by said stub-shaft, said motor comprising two cylinders arranged obliquely to each other, connecting rods, a crank-disk, connections between said crank disk and said wheel, and means for driving said motor.

October 15, 1907.

868,441. COMPRESSED-AIR MOTOR. CHARLES D. JENKINS, Boston, Mass. Filed May 6, 1907. Serial No. 372,117.

1. The combination with a source of air under pressure, of an air engine comprising a cylinder and a piston reciprocating in said cylinder and coöperating therewith to form a substantially small chamber when the piston is at the end of its stroke in one direction, means to admit compressed-air into said chamber, an electric conductor located in said chamber when the piston is at the end of the stroke on which said chamber is formed and capable of being highly heated, an electric circuit in which said conductor is included when the piston is at the end of its chamber-forming stroke, and a circuit controller for said circuit operated to close the same and highly heat the conductor when the piston is at or near the end of its chamber-forming stroke to thereby expand the compresed-air and propel the piston, substantially as described.

2. The combination with a cylinder and a piston reciprocating therein to form a substantially small chamber when the piston is at the end of its stroke in one direction, an electric conductor movable with said piston, an electric circuit having terminals extended within said cylinder and coöperating with said electric conductor to include the latter in a closed circuit and thereby highly heat the



conductor when the piston is at the end of its stroke in one direction, substantially as described.

GERMAN STATE RAILWAYS - DIESEL-COMPRESSED AIR LOCOMOTIVE

(For description see opposite page.)

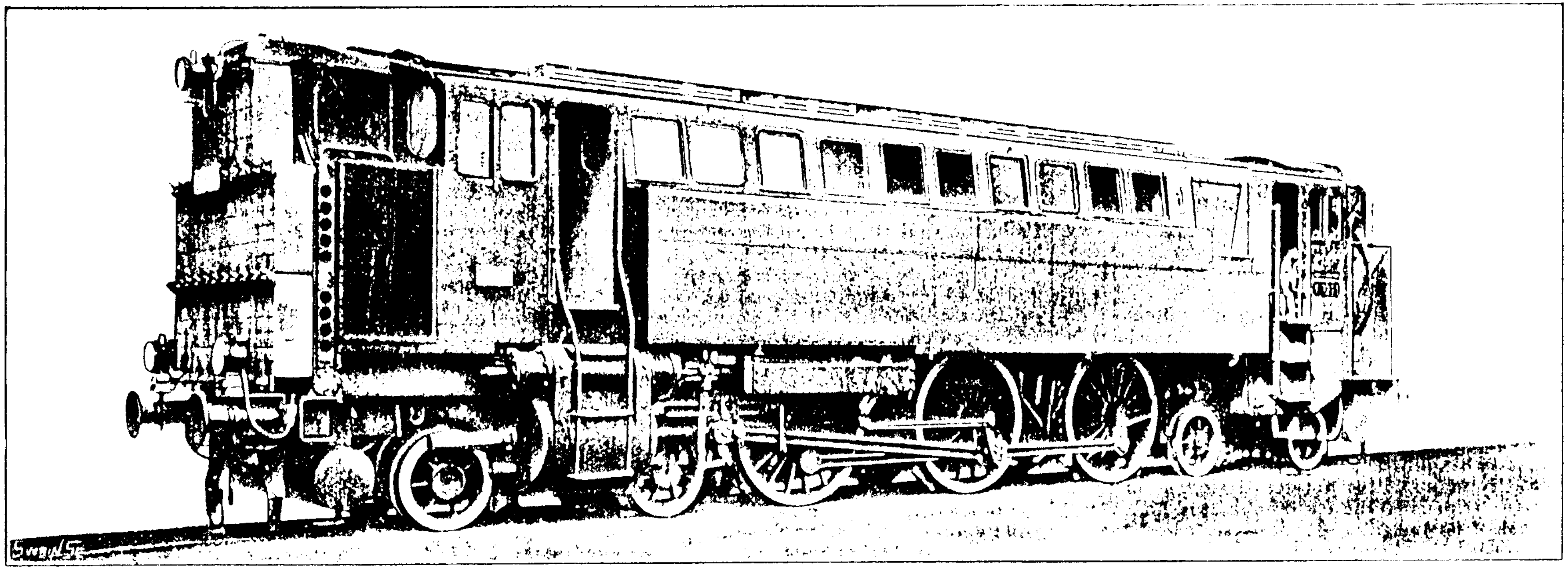


FIG. 1—GENERAL VIEW OF THE LOCOMOTIVE

1200 B.H.P. Diesel-Compressed Air Locomotive for the German State Railways.

IN our annual review of locomotive progress during 1929, which appeared in THE ENGINEER for January 3rd, 1930, brief mention was made of the new Diesel-compressed air locomotive, which has been designed and built by the Maschinenfabrik Augsburg Nürnberg A.G. and the Maschinenfabrik Esslingen for the German State Railway Co. A view of the completed locomotive is reproduced on page 486, and it may be recalled that in the official tests carried out in the Stuttgart district in November last, the engine pulled a train weighing 233 tons up the Geislinger Steige gradient of 1 in 43 mostly on curves, at an average speed of over 20 kiloms. per hour.

Some particulars of this locomotive were given in the *V.D.I. Journal*, No. 10, of March 8th last, in an article by Reichsbahnrat Witte and Mr. R. P. Wagner, while in the same *Journal*, No. 12, of March 22nd, Dr. Ing. Jos Geiger, of Augsburg, contributed a further article entitled "Diesel Locomotive with Compressed Air Transmission."

The article which follows has been prepared from some further notes by Dr. Geiger, which have been placed at our disposal by the London representatives of the M.A.N. Company—John Le Boutillier, Ltd., of 13, Rood-lane, E.C. 3. The locomotive is now at the Grunewald railway shops of the Reichsbahn, and very thorough tests are being carried out. It is hoped that an account of the test performance will be given in one of the papers to be read at the forthcoming Power Conference, which is to be held from June 16th to the 25th in Berlin.

At the time the locomotive was ordered, the following position influenced the builders and the railway officials in their decision to adopt compressed air transmission. Up to the time of the placing of the order experience had already been gained with internal combustion locomotives embodying the direct drive, the geared drive, the fluid transmission drive, and the electric drive, while locomotives embodying these different systems had either been built and tested or were then in course of construction. At that time, however, a locomotive of the internal combustion type operating with compressed air transmission had not been built. On many sides the opinion prevailed that such a method of power transmission was valuable on several grounds. Thus, the hyperbolic curve of tractive power for compressed air working approached very closely that for steam and fitted in well. Again, by utilising compressed air, difficult manœuvring operations were avoided, and the driver had only to attend to the regulation of the supply of fuel to the oil engine. Another advantage of the compressed air system, it was held, was that the driver would have to deal with essential parts of the locomotive already familiar to him in steam practice.

On the other hand, there were those who foresaw difficulties, and did not hesitate to prophesy failure. It was said that the efficiency of the compressed air system of power transmission was lower than that of any other system, and it was also suggested that explosions arising from oil in the compressed air might take place, while stoppages could occur owing to the formation of ice on the cylinders in which the air was to be expanded.*

* See *Diesel Power* for February, 1930, page 94.

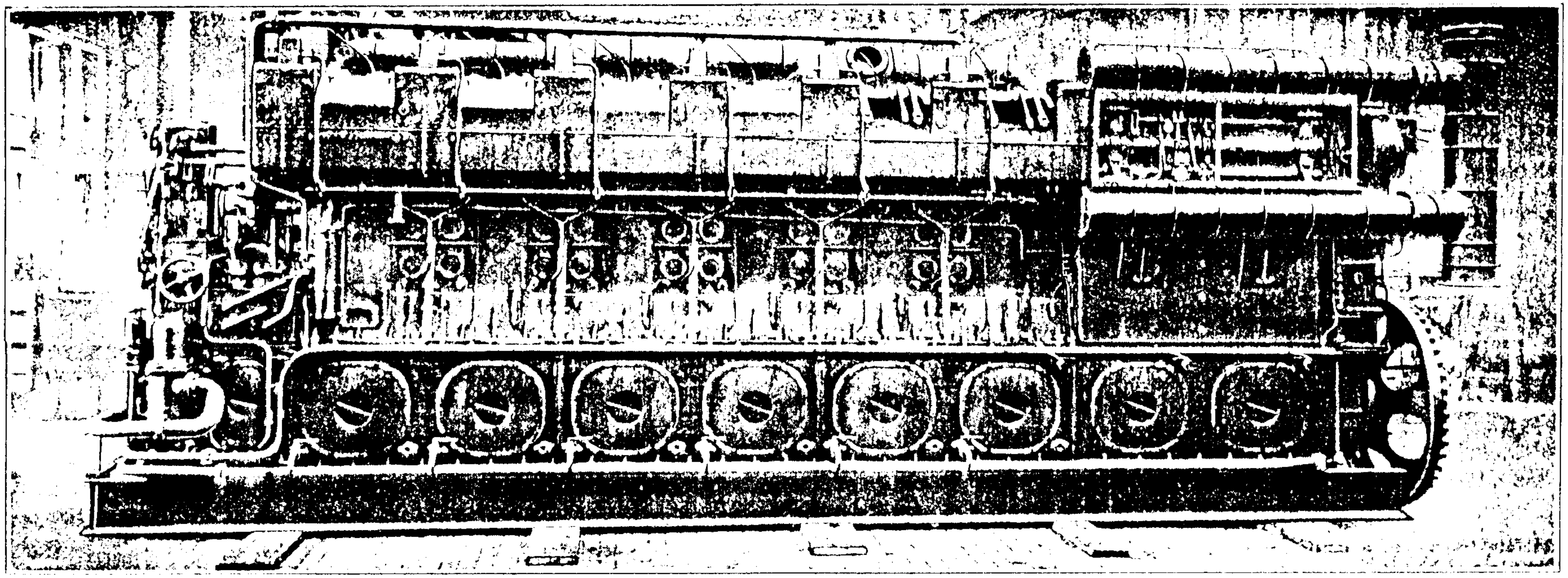


FIG. 2—M.A.N. 1200 B.H.P. DIESEL ENGINE AND AIR-COMPRESSOR

Such statements were made even after actual experience with the locomotive showed that such happenings did not occur. We are informed that, in fact, the efficiency of transmission with the oil-compressed air system has been demonstrated to be higher than with the oil-electric system, enabling a 12 per cent. saving on the fuel used per ton-kilometre to be recorded at a common speed of 60 kiloms. per hour. There have been no explosions of oily vapour, even although in a works test it was sought to produce such explosions artificially by the introduction of electric sparks.

During the long series of trials which have been carried out, there was at no time any signs of ice formation on the cylinders of the locomotive. On the contrary, when the hand was placed in the air at the exhaust outlet it was quickly withdrawn, since the air, after being expanded to close upon atmospheric pressure, still retained a temperature of about 100 deg. Cent. No difficulties have been experienced with the deposit of dirt from particles of lubricating oil or carbon in the air reheater, or with the clogging of the air compressor valves from like causes. A photograph of two of the air compressor valves which were removed after the completion of all trials is reproduced in Fig. 5. It clearly indicates the comparatively clean state of the valves.

General Particulars.

In the following table the general particulars of the locomotive and the leading dimensions of the oil engine and air compressor are given. In Fig. 1, on page 486, a general view of the completed locomotive is reproduced, while in Figs. 2 and 3 views of the engine and air compressor on the test bed at the makers' works are given.

Locomotive Particulars.

Type	4-6-4
Tractive effort at driving wheel rims	12,000 kilos.
Diameter of driving wheels	1600 mm.
Diameter of locomotive cylinders	700 mm.
Stroke of locomotive cylinders	700 mm.
Adhesive weight	54,600 kilos.
Weight empty	118,600 kilos.
Weight in working order	124,000 kilos.
Weight of fuel oil carried	2,000 kilos.
Maximum designed speed	80 kilom. per hr.

Oil Engine and Air Compressor Particulars.

Type of engine	M.A.N. four-stroke single-acting	
Number of cylinders	Six	
Bore of cylinders	450 mm.	
Stroke	420 mm.	
	Continuous load.	Short overload.
Designed B.H.P. output	1000	1200
Designed I.H.P. output	1350	1630
Revolutions per minute	400	450
Working air pressure	6·5 atm. (92·3 lb. per sq. in.) to 7 atm. (99·4 lb. per sq. in.)	
Temperature of air at locomotive cylinders	330 deg. to 360 deg. Cent.	
Type of air compressor	Twin-cylinder single-stage double-acting	
Diameter of cylinders	640 mm.	
Stroke	350 mm.	

The oil engine, as shown by our illustration, is of the normal vertical cylinder type and is generally similar to that supplied for the Russian Diesel electric locomotives, but embodying several improvements in design. The bed-plate is common to the engine and air compressor, which are direct-coupled. A section through one of the compressor cylinders, which we reproduce in Fig. 4, shows the mechanically operated suction valves and the plate-type delivery valves.

Dr. Geiger claims that the speed of the compressor is remarkable since up to the present air compressors dealing with as much as 182 kilograms of air per minute, have only been built for speeds up to 100 r.p.m. The overall efficiency, we learn, is very high, in spite of the large range of compression, in the single stage cylinders. It was found that equal values were obtained when the volume of air delivered per minute as ascertained by nozzle tests was compared with the calculated swept volume after allowing for re-expansion in the clearance spaces, which fact is itself a good indication of the very small loss by throttling in the suction and delivery of the air through the valves. The compressed air on leaving the compressor passes directly to a reheater, which operates on the contra-flow principle, and by utilising the heat of the exhaust gases raises the temperature of the air to 360 deg. Cent. From the reheater the air passes through the regulator control valve to the two cylinders of the locomotive in which it is expanded to practically atmospheric pressure.

From the illustrations reproduced on page 486, it will be noted that laminated coolers of the fully welded type are arranged at both ends of the locomotive. These coolers serve the circulating water system for the engine jackets and bearings, the piston cooling oil circuits, and the lubricating oil circuit. The compressor is cooled by the [illegible]

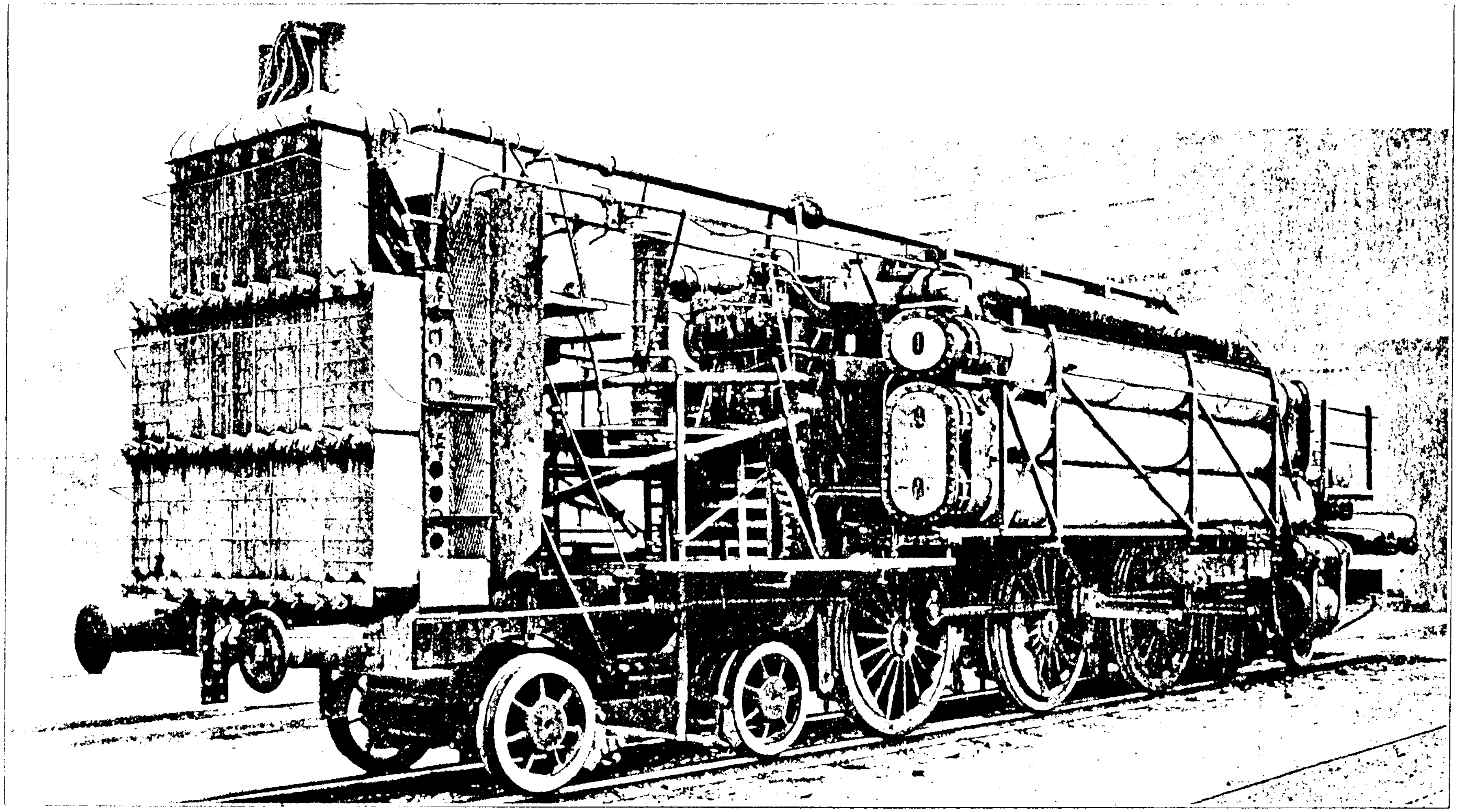


FIG. 3—VIEW SHOWING THE AIR-COMPRESSOR AND ACCESSORIES

injection of water, which will be referred to later.

Under normal conditions, the natural draught through the coolers is sufficient, but electrically driven fans are provided to meet exceptional cases.

A small direct-driven dynamo provides current for the fan motors and for train lighting, and the compressed air for the brakes is taken directly from the main compressor. The only other auxiliary appliance is an oil-fired boiler for train heating. There are two driving cabs, so that the locomotive can proceed in either direction without the necessity of turning.

Two essentials of the compressed air system of transmission are the injection of water into the compressor for the purpose of cooling the air, and the reheating of the air by the exhaust gases. As will be seen from Fig. 1, the air compressor has neither cooled jackets nor cooled covers. The air is cooled by injecting water in the form of a fine mist into the compression space. For this purpose a special pump with a particular kind of spray nozzle is employed. By the use of water injection it is possible so to cool the air that the compression curve approaches the isothermal line very closely. This could not well be done with surface cooling, especially in the high-speed type of compressor employed. The injection of water into the cylinders also adds weight to the working medium and allows a very much lower entry temperature at the reheater to be employed than would otherwise be possible. This lower entry temperature also enables the heat of the exhaust gases to be better utilised. With the comparatively low temperature of the air leaving the compressor, the delivery valves are rendered more reliable in service and any danger of explosion by the ignition of oil particles is entirely overcome.

THE RUNNING OPERATIONS.

The running of this Diesel-compressed air locomotive is very simple. Immediately before setting out, the oil engine is started up and the air compressor run light with no air circulating through the reheater. As soon as the regulator lever is moved the air compressor is automatically put on load and the pressure of the air delivered rapidly rises so that the locomotive starts as easily as with steam. When running the supply of air to the cylinders is reduced to that necessary for the speed and further adjustments in speed are then made by varying the amount of fuel delivered to the cylinders of the oil engine. When more fuel is admitted the engine increases its speed and more air is delivered by the compressor, with the result that the air pressure rises and increases the speed of the locomotive. The full supply of air is, however, only given to the cylinders when going up the steepest gradients. The driver can also vary the amount of water injected into the air compressor cylinders, but this requires little attention and he can give his full time to watching the track and signals.

It is said that the cost of upkeep is small, and it is held that the normal repair and maintenance facilities as arranged for steam locomotives will suffice. Although some water is required for injecting into the air compressor the amount even under full load conditions is only 1 per cent. of that which would be required by a steam locomo-

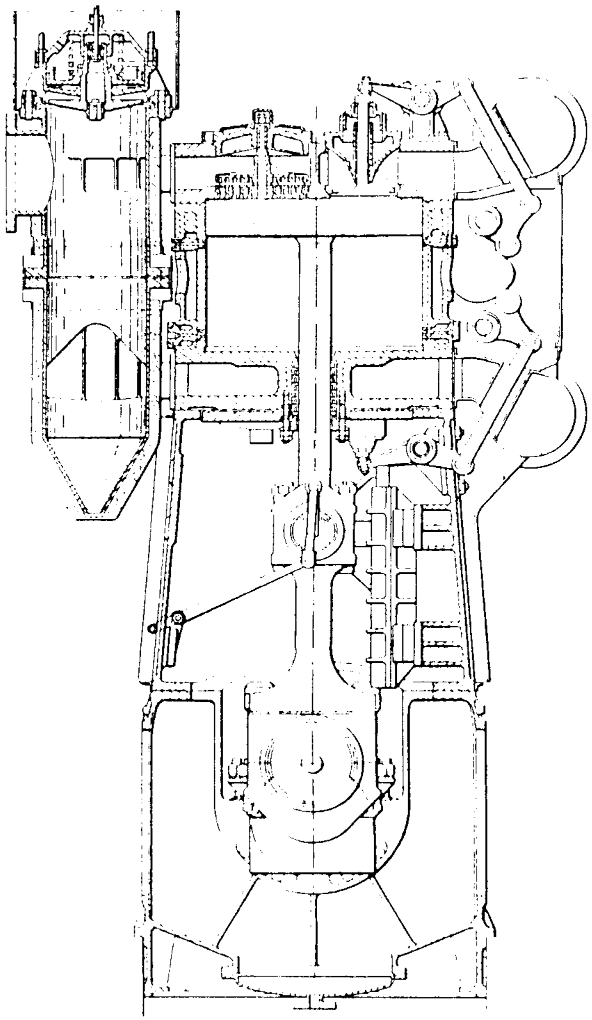


FIG. 4- AIR - COMPRESSOR

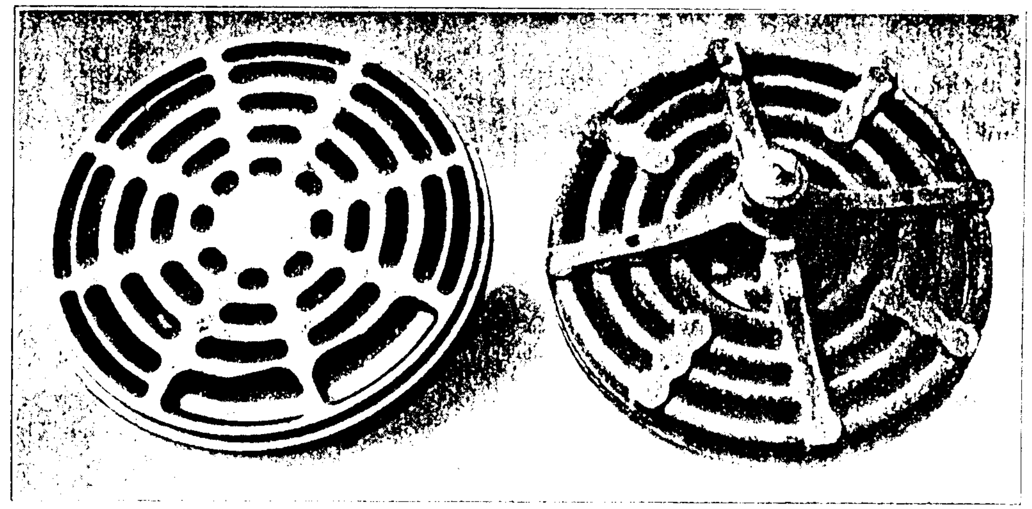
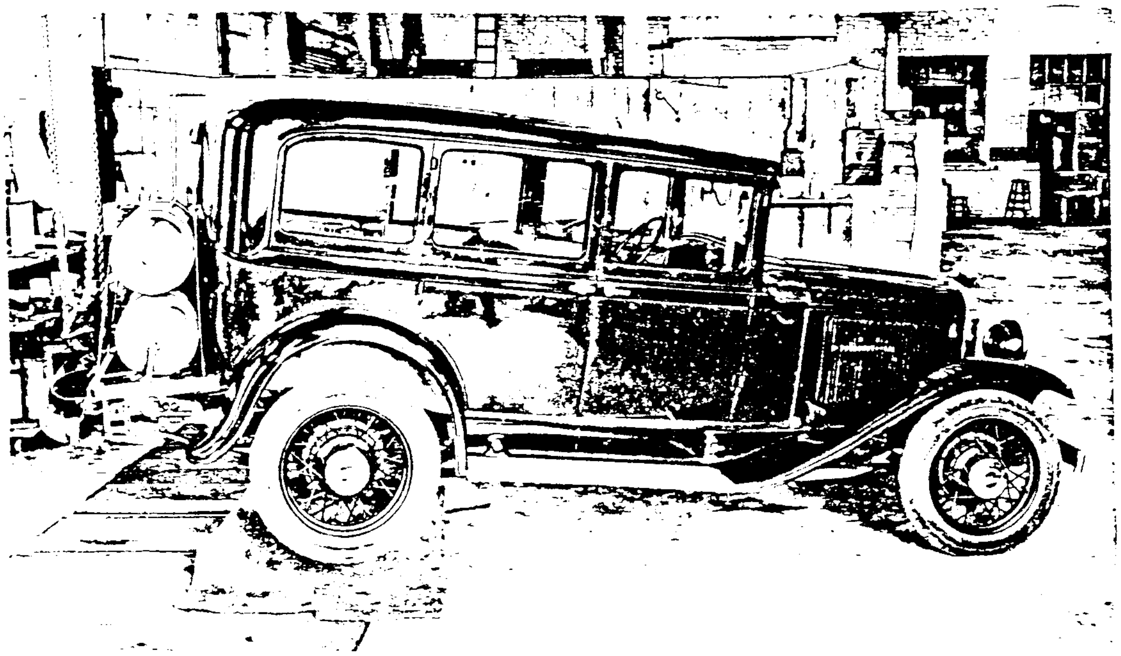


FIG. 5 -AIR - COMPRESSOR VALVES

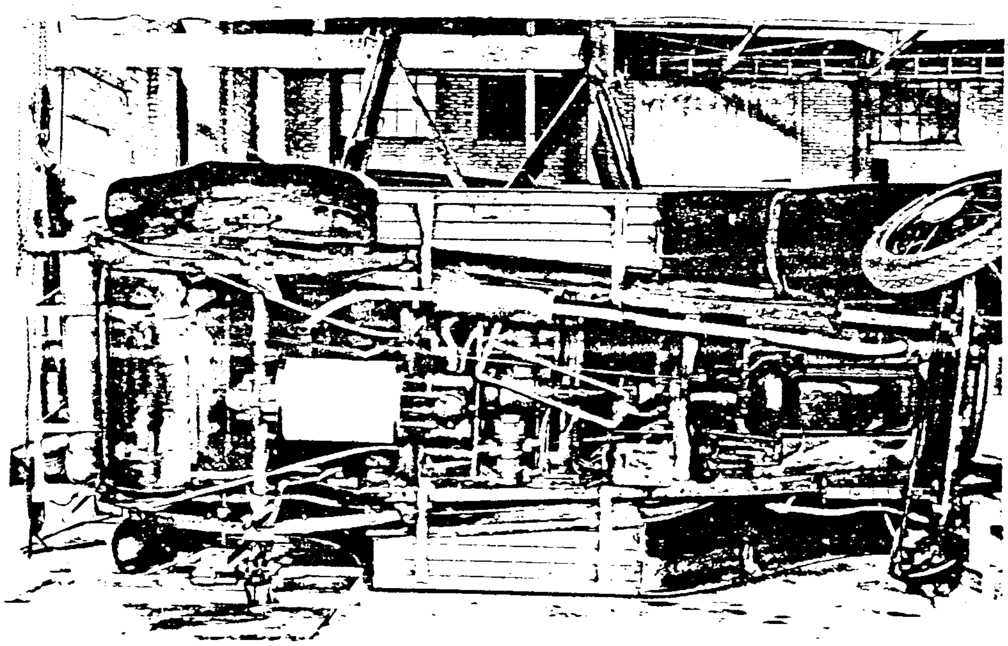
tive, and at light loads no water is required. Further details of the test performance of this new locomotive, which it is hoped will be published by the Grunewald railway testing department, will be awaited with interest.

1932 First Built Air Drive Automobile Plymouth

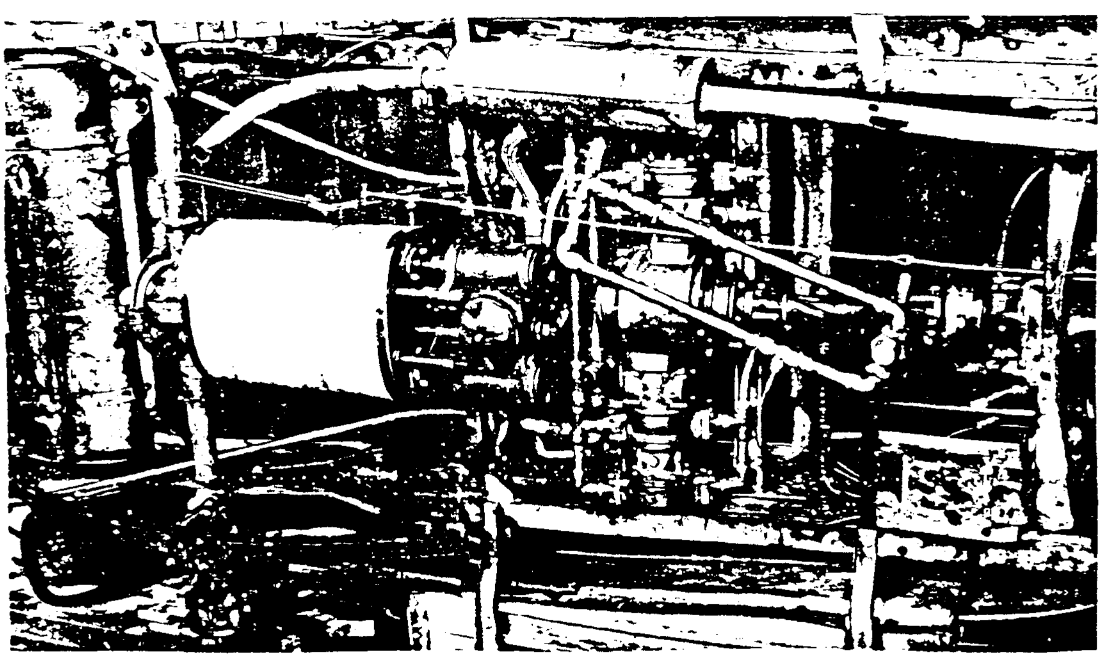


Plymouth retro-fitted with Stanley Steamer engine used as air motor

Air storage tanks are shown at the rear



Under side of Plymouth showing original engine at right as prime mover, and the Stanley Steamer engine at left used as air motor to drive the vehicle



More detail of the under side of 1932 Plymouth showing the Stanley mechanism at left used as air motor, and showing the compressor in the center

San Marino Tribune (and San Marino News)

Vol. 51 No. 1 282-5707 792-3343 San Marino, California Thursday, January 3, 1980

Air Compression Moves Auto

Burt's Kit Converts Auto

Dr. Robert Burt

Dr. Robert Burt of San Marino has an idea which could revolutionize the auto industry— a kit, installed in any auto, that would transform the engine from a gas-eater to an air-user. Burt says it's a matter of "changing a car's appetite."

Actually this isn't just an idea. It's an invention he perfected in 1932 and he has updated it and plans to utilize this "Burt Retrofit Kit" in his 1966 Cadillac. Originally he called his invention the "Burt Air Drive."

"There is more energy in one gallon of gasoline than in 20 pounds of dynamite," says Burt. That energy only moves a Cadillac 10 miles proving that gasoline is a "very inefficient energy."

Simplified, Burt says his idea uses the steam engine technique but makes use of compressed air instead of steam.

How does it work? Burt says it's really very simple.

Its principle is the same as a locomotive except that the "works" are underneath the auto. He adds that the motor runs by taking air out of the storage area (containing about 250 pounds) and pumps it up to three times that pressure.

The engine-compressor unit replaces the boiler and fire box of the steam car, and the clutch, gearbox and propeller shaft of conventional cars.

Burt says the engine is more efficient because "it is like having a 10-speed overdrive. The engine is never used to slow or stop the car; the air motors are reversed and energy is saved. What cannot be stored is by-passed through a safety valve and passed through the radiator.

"The air, after compression, is used to cool the engine cylinders and is also super-heated in the exhaust system, therefore recovering much power from wasted heat."

The kit Burt proposes for general marketing includes a small standard four-cylinder engine which is really the prime mover and an air compressor attached to that engine. The air flows into a high pressure tank resembling a scuba bottle for storage of energy, and an air valve performs like a throttle controlling the air from the storage tank to the motor. The fire wall is directly connected to the drive shaft. Simply open the throttle and air drives the motor that moves the car, he says.

It will work easily in any auto because various kits will be designed to fit and bolt onto the existing auto frame, i.e., a certain numbered kit will fit uniformly all autos of a certain size. There will be kits to fit onto all autos, he says.

The reason why Burt knows his idea is usable is because in 1932 he converted a Plymouth and it ran. Then his idea echoed back to him in 1973 when gas lines started forming.

Burt says, "If we could cut gasoline consumption by 75%, we would also cut the pollution by that percentage. I think we can cut it (consumption) by 95%."

In Dr. Burt's words, "The auto is the most sophisticated and useful form of transportation the world has ever known."

But Burt doesn't reserve his talents to improving only the auto. He is the inventor of the photo-electric cell which he describes as "the radio tube that receives light and controls electric current by the action of light." He also has a patent on the first cathode ray oscilloscope and the bone-conduction hearing aid later taken over by commercial organizations. Another of his developments was the Beltone talking picture theater equipment. While serving at Lockheed he a won a $100 war bond for the best production short-cut invention of the month— anti-glare welders' goggles which he designed in collaboration with the Corning Glass Works.

Burt has always been a natural when it comes to inventions. At the age of 10 he had already started a business as "doorbell doctor" in his home town, Battle Creek, Michigan. He often ended up fixing other electric gadgets in the homes he visited and soon invented a glider which had everything present-day models have except towing devices to get it into the air.

After graduating from Cornell University in 1921, he continued his studies and received his Ph.D. in physics from Caltech. Failing to get into WW I because of an injury due to an auto accident, he did research work for General Electric and Western Electric, before getting his degree at Caltech.

Let's hope that in about 18 months— Burt's prediction for manufacture of the auto kit— all of us will be able to buy the Burt Retrofit Kit and transform all our cars into air-users instead of gas-guzzlers.

Burt Energy Research Corp.
A Non-profit Foundation
544 Bonita Ave.
San Marino, CA 91108

June 14, 1938. R. C. BURT 2,120,546

MECHANICAL TRANSMISSION

Filed May 25, 1931 2 Sheets-Sheet 1

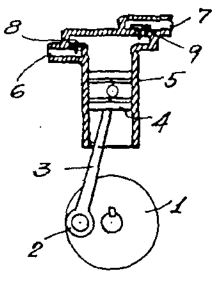


Fig 1.

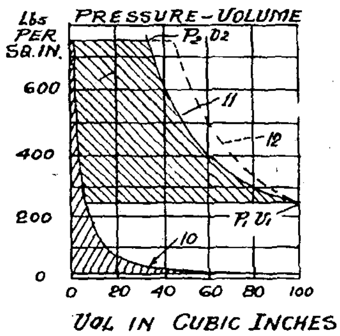


Fig. 2.

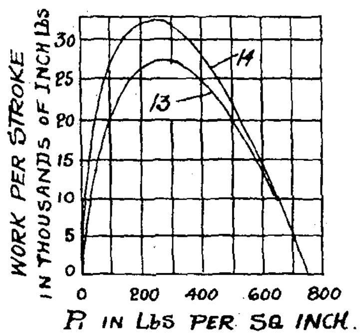


Fig 3.

Fig. 5.

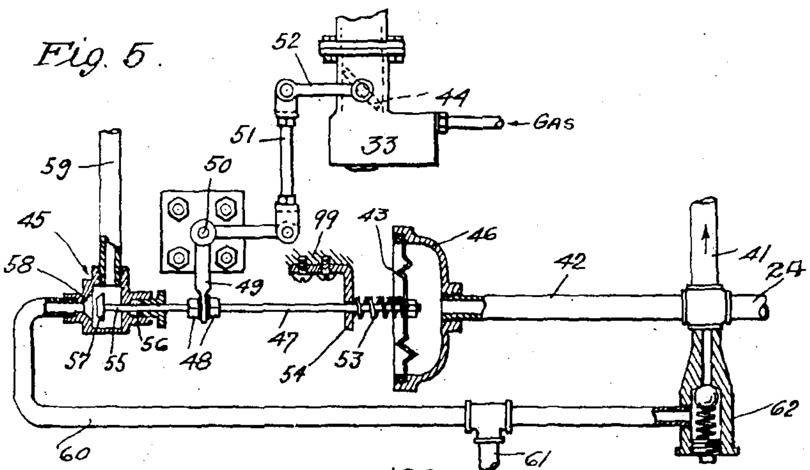
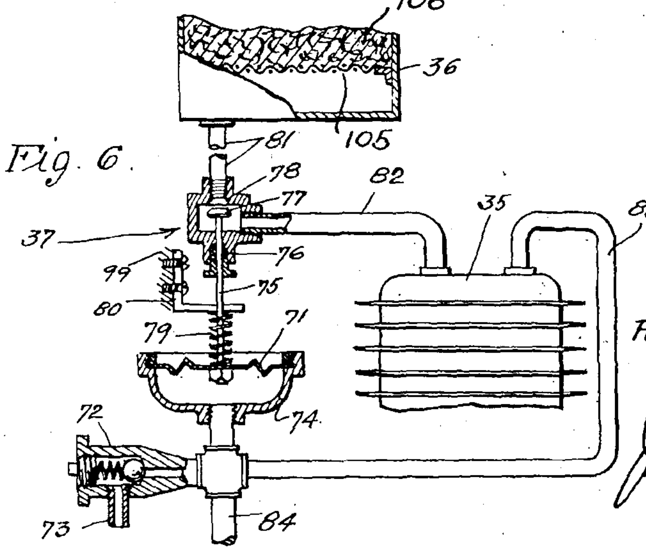


Fig. 6.



Inventor
Robert Cady Burt.

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June 14, 1938. R. C. BURT 2,120,546

MECHANICAL TRANSMISSION

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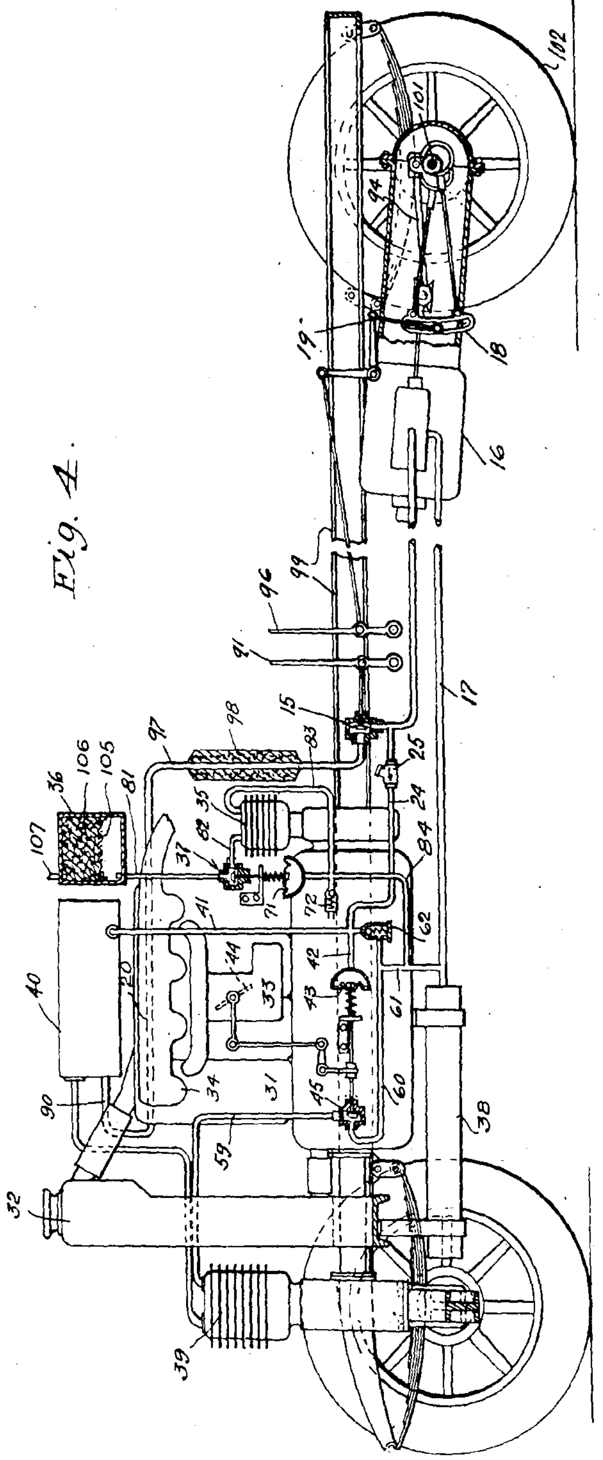


Fig. 4.

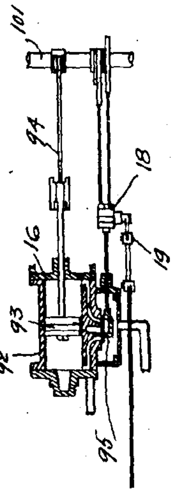


Fig. 7.

Inventor
Robert Cady Burt.

Attorney

Patented June 14, 1938

2,120,546

UNITED STATES PATENT OFFICE

2,120,546

MECHANICAL TRANSMISSION

Robert Cady Burt, Pasadena, Calif.

Application May 25, 1931, Serial No. 539,966

2 Claims. (Cl. 60—14)

This invention relates to apparatus for the transmission and control of mechanical energy, force, power, or effect, from a source to any desired point.

This invention has for its object the economical transmission of mechanical power, with great flexibility of power and speed and fine control.

This transmission of power is accomplished through the medium of an elastic gas, vapor, or fluid.

In the following disclosure and claims the term "fluid" is used broadly to denote gas, vapor, air, or compressible liquid, or other material of like characteristics.

A compressor or pump is attached to the source of power and this compressor takes the fluid at a pressure P_1 and compresses it to a higher pressure P_2, thus doing mechanical work on the fluid and storing in the fluid the energy of compression. Another mechanism, known as a motor, takes this fluid and expands it back from pressure P_2 to P_1, thus extracting the power stored in the fluid during compression.

For many uses of this transmission it is desirable to use compressed air for the working substance or elastic fluid, and in many places in this disclosure I shall refer to the elastic fluid as the air, but I do not limit my invention to a transmission using air. In fact, under certain conditions, other vapors appear to be more desirable.

Referring to the drawings: Fig. 1 is a sectional view of an air compressor; Fig. 2 is a pressure volume diagram of an air compressor; Fig. 3 is a graph showing the variation in work done by an air compressor when the initial volume and final pressure are held constant and the initial pressure is varied; Fig. 4 illustrates one specific application of this transmission to a motor car; Fig. 5 is a diagram of the high pressure control system; Fig. 6 is a diagram of the low pressure control system; Fig. 7 is a horizontal longitudinal sectional view of the air motor.

In Fig. 1 is diagrammed an ordinary air compressor, having a crank shaft 1, a crank 2, connecting rod 3, piston 4, which moves up and down with the rotation of the crank-shaft inside of the cylinder 5 which the piston tightly fits. On the down stroke of the piston air is drawn into the cylinder through the intake 6 and inlet valve 8 and is compressed on the up stroke of the piston, being forced out of the compressor through check valve 9 and exhaust outlet 7. The amount of compression depends upon the intake pressure P_1 and the pressure into which the exhaust is discharging or P_2.

It is well known to engineers that the work performed by the piston on the gas in this operation is equal to the integral of the pressure multiplied by elemental changes in volume taken around the cycle.

To illustrate:—assume we have a cylinder of 100 cubic inches capacity, that it is filled with air at 15 lbs. per sq. inch and that no air can escape until we have compressed this air to a pressure of 750 lbs. per sq. inch. Curve 10 of Fig. 2 shows how the pressure within this cylinder increases as the volume is decreased until 750 lbs. per sq. inch pressure is reached and then the air is exhausted. The statement in the preceding paragraph simply states that the work done by the cylinder on the air is proportional to the area to the left and below the curve 10 and above the horizontal line of 15 lbs. per sq. inch and to the right of the vertical line representing zero volume.

In the formulae employed to explain my invention, the following characters appear and have the meanings here set forth:

P_1=pressure of fluid prior to compression.
P_2=pressure of fluid after compression.
P or p=pressure of a fluid, generally.
V or v=volume of a fluid, generally.
T=temperature of a fluid, generally.
K=gas constant of a fluid, generally.
C_p=specific heat of a fluid at constant pressure.
C_v=specific heat of a fluid at constant volume.
W=work of compression of a fluid.
e=the logarithmic exponent =2.718.
K^1=a constant related to K.
V_1=volume of fluid prior to compression.
$\gamma=C_p$ divided by C_v.

If the compression has been performed isothermally or at constant temperature, the equation of the curve is given by Equation 1, as follows:

$$PV=KT \qquad (1)$$

If the compression has taken place adiabatically or without loss of any heat by the air, then the

equation is that given by Equation 2, as follows:

$$PV^{\gamma}=\text{Constant} \qquad (2)$$

Curve 11, Fig. 2 represents the same equation (No. 1 above) as curve 10, except that the initial pressure P_1 is higher and the work done is represented by area bounded by the lines $P_1=250$ lbs. per sq. inch; $v=$zero; $P_2=750$ lbs. per sq. inch; and the curve 11. Curve 12 represents the Equation 2 above, at the same volume and pressure as curve 11.

$$\frac{Cp}{Cv}=\gamma \qquad (3)$$

Equation 3 states the well known gas law:—specific heat at constant pressure C_p, divided by specific heat at constant volume C_v, is a constant. This constant is represented by the Greek letter γ. For air it is 1.4.

To obtain the expression for the work of compression at constant temperature, we proceed as follows:

$$\text{Work (T constant)}=W=\int p\,dv$$

Or, instead of integrating pdv around the cycle, we may integrate vdp between the limits of P_1 and P_2, which is the same thing. Thus,

$$W=\int_{P_1}^{P_2} v\,dp=P_1V_1\int_{P_1}^{P_2}\frac{dp}{p}=P_1V_1\Big[\log_e p\Big]_{P_1}^{P_2}=$$

$$P_1V_1\log_e\left(\frac{P_2}{P_1}\right) \qquad (4)$$

The expression for the work of adiabatic compression is derived as follows:

$$PV^{\gamma}=K, \text{ or } P^{\frac{1}{\gamma}}V=K^1$$

$$W=\int_{P_1}^{P_2} v\,dp=P_1^{\frac{1}{\gamma}}V_1\int_{P_1}^{P_2}\frac{dp}{p^{\frac{1}{\gamma}}}$$

$$=\frac{P_1^{\frac{1}{\gamma}}V_1}{1-\frac{1}{\gamma}}\left(P_2^{\,-\frac{1}{\gamma}}-P_1^{1-\frac{1}{\gamma}}\right) \qquad (5)$$

If the initial volume V_1 and final pressure P_2 are held constant, and if different values of the input pressure P_1 are taken, it will be found that the work done by the compressor increases as P_1 increases up to a certain value and then decreases as P_1 approaches P_2 in value. This is shown graphically as an example in Fig. 3 where V_1 is taken as 100 cu. inches; P_2 is taken as 750 lbs. per sq. inch; and the work W is plotted against the different values of P_1. Curve 13, Fig. 3 corresponds to Equation 4 and curve 14 corresponds to Equation 5. Both curves have maxima at nearly the same value of compression ratio. These maxima are obtained by differentiation of Equations 4 and 5, obtaining 6 and 7 below.

Differentiating Equation 4, holding P_2 constant, we have

$$P_1V_1\log_e\left(\frac{P_2}{P_1}\right)=P_1V_1(\log_e P_2-\log_e P_1)$$

$$\frac{dW}{dP_1}=(\log_e P_2-\log_e P_1)V_1-\frac{P_1V_1}{P_1}=\left(\log_e\frac{P_2}{P_1}-1\right)V_1$$

Equating the derivative

$$\frac{dW}{dP_1}$$

to zero to find the maximum value for W, results in a maximum value appearing when

$$\frac{P_2}{P_1}=e=2.718 \qquad (6)$$

Differentiating Equation 5, holding P_2 constant, we have:

$$W=\frac{P_1^{\frac{1}{\gamma}}V_1}{1-\frac{1}{\gamma}}\left(P_2^{1-\frac{1}{\gamma}}-P_1^{1-\frac{1}{\gamma}}\right)=\frac{V_1}{1-\frac{1}{\gamma}}\left(P_1^{\frac{1}{\gamma}}P_2^{1-\frac{1}{\gamma}}-P_1\right)$$

$$\frac{dW}{dP_1}=\left(\frac{V_1}{1-\frac{1}{\gamma}}\right)\left(P_2^{1-\frac{1}{\gamma}}\frac{P_1^{\frac{1}{\gamma}-1}}{\gamma}-1\right)$$

By equating to zero, we find a maximum value for W when

$$\frac{P_2}{P_1}=\gamma^{\frac{\gamma}{\gamma-1}} \qquad (7)$$

For air, $\gamma=1.4$, and

$$\frac{P_2}{P_1}$$

becomes 3.2 for maximum work.

It is also evident from Fig. 3 that much more work can be obtained from a given cylinder working at a limited upper pressure P_2 by proper selection of the initial pressure P_1. For example, the compressor diagrammed in Fig. 2 would absorb less than 6000 inch lbs. per stroke when operating from atmospheric pressure to 750 lbs. per sq. inch as in curve 10 and it would absorb more than 27,000 inch lbs. per stroke when operating from 250 lbs. per sq. inch to the same upper pressure.

From the foregoing consideration, it is evident that a power transmission can be designed using previously compressed air for its low pressure intake and compressing it over a comparatively small pressure ratio to a higher pressure. Then, after transmission through a pipe and control, the compressed air can be expanded through an air motor back to the same pressure as that from which it started.

By this means a transmission of extreme lightness, flexibility, compactness and economy and having other desirable features, may be obtained.

Fig. 4 shows a specific application of this transmission to a motor car and, while I do not limit my invention to automobile transmissions, it will serve as an example to illustrate the principles involved.

Referring to Fig. 4, 31 is a gasoline engine having a radiator 32, carburetor 33, and exhaust manifold 34. This engine, being started in any of the usual manners, drives a small auxiliary compressor 35. This compressor takes air from the atmosphere through a cleaner, conventionally illustrated at 36, and throttle 37, compresses and discharges it through the pipe shown into tank 38. The cleaner 36 may consist of a box or can having supported, spaced from the bottom, a wire screen 105 above which is placed bronze wool 106. The intake pipe 107, open to the atmosphere, enters at the top, and the discharge pipe 81 passes out the bottom of the cleaner. At the same time the power compressor 39 has been taking air from 38, compressing and discharging it into tank 40. This process is continued until the pressure in tank 40 has been built up to a certain predetermined value, for example, 750 lbs. per sq. inch. When P_2, the pressure in tank 40, has reached this pressure the connecting tube 41 and the tube 42 transmits this pressure to the diaphragm 43 which expands

closing the throttle 44 of carburetor 33 and closing the intake to pump 39 by action of valve 45.

The action of this control mechanism can be best understood by referring to Fig. 5. Diaphragm 43 is secured at its outer edge to shell 46 which is mounted on the car frame 99. Attached to the center of diaphragm 43 is a rod 47 on which are nuts 48 spaced somewhat apart. Carried between these nuts is bell crank 49 pivoted about a center 50 and operating on the butterfly valve 44 of carburetor 33 through links 51 and 52. The movement of diaphragm 43 due to pressure in shell 46 is resisted by springs 53 which is in compression between the diaphragm and bracket 54 mounted on the car frame. An extension 55 of rod 47 passes through packing 56 and carries a cone 57 that seats against seat 58 of valve 45 when diaphragm 43 is extended. Compressor 39 takes in air through pipe 59, valve 45, pipe 60, pipe 61, from the low pressure tank 38. It is obvious, then, that the action of the diaphragm 43 under pressure is to close the engine throttle 44 and also the valve 45, choking off the intake to the compressor and unloading it.

At this stage the engine idles under little or no load and the pressures are maintained. When P_1 reaches its proper value, for example, 250 lbs. per sq. inch, then diaphragm 71 closes valve 37 and no more air is pumped into the system until some escapes. Should pressure P_1 become too high, safety valve 72 lets air escape to atmosphere through pipe 73. In the same manner safety valve 62, allows air to escape from tank 40 through pipes 41, 60, and 61 back to low pressure tank 38, if P_2 should become too high.

The action of the low pressure control can be seen in detail by referring to Fig. 6. Diaphragm 71 is secured at its outer edge to shell 74 which is mounted on the car frame. Attached to the center of the diaphragm is rod 75 that extends through packing 76 and carries a cone 77 that seats on valve seat 78 of valve 37 when diaphragm 71 is extended. The movement of the diaphragm is resisted by spring 79 which is in compression between diaphragm 71 and bracket 80 mounted on the car frame. Compressor 35 takes air from the atmosphere through cleaner 36, pipe 81, valve 37, and pipe 82, discharging it through pipe 83, and pipes 84 and 61 into low pressure tank 38. When the pressure in tank 38 reaches a predetermined value the diaphragm 71 will have extended, pushing the rod 75 upward and closing the valve 37. The compressor 35 is then unloaded and inactive until the pressure in tank 38 drops sufficiently to permit the diaphragm actuated valve to open and allow air to be taken in by the compressor.

High pressure air from tank 40 is brought back through pipe 90, through throttle valve 15, operated by lever 91, to the intake side of engine 16 which expands the air and returns it to tank 38 through pipe 17. Thus no air is lost from the system.

Engine 16 is exactly like a steam or compressed air engine having the usual cylinder 92, piston 93, connecting rod 94, and valve mechanism 18 which through the valve link 19 controls the time of cut-off and admission as well as the exhaust events through change of the phase relation of valve 95 with respect to piston 93. A standard Stephenson link has been shown and since this valve gear is so old and so well known to those skilled in the art, it is not considered necessary to explain its operation in detail. The connecting rod 94 operates through a crank to turn rear axle 101 to which are attached driving wheels 102.

The entire mechanism up to throttle 15 and return pipe 17 is automatic, the entire control of the automobile is accomplished by manipulation of the throttle 15 through lever 91 and control 19 through lever 96. These are operated exactly in the same manner as a steam engine.

Many economies may be incorporated in this system. For example: compressor 39 and tank 38 may be placed out in front where, being in the air stream incident to the travel of the vehicle, they would be cooled, thus reducing the volume of air to be compressed, and the compressor 39 and tank 38 are so shown in this preferred arrangement in Fig. 4. The high pressure air may be passed through a heat interchanger 20 in the exhaust manifold for the purpose of expanding the air, or reducing its density, by taking waste heat from the exhaust gases. The pipe 97 and engine 16 may be heat insulated to conserve this heat.

Another economy of the system is that the gasoline engine is always working under high torque and never running at high speed and light load. As will be understood from the control operations explained, the engine is always either operating with full compressor load, or is idling at closed throttle with the compressor unloaded.

A simple pipe 24 and check valve 25 shunting valve 15 permits regenerative braking. For regenerative braking the valve 15 is closed and the links 19 are gradually shifted to the reverse position, thus causing air to be taken from tank 38, compressed by engine 16 and driven through pipe 24, check valve 25 and pipe 41 back into tank 40. This continues until a higher pressure is reached in tank 40, of, perhaps, 1200 lbs. per sq. inch. Thereafter air escapes through relief valve 62 into tank 38 again with a small amount of heating, due to the Thompson-Joule effect.

After this regenerative braking process, the excess pressure is available for subsequent acceleration without any call upon the motor 31.

While I have shown reciprocating motors and compressors, I do not wish the invention in certain of its aspects to be limited thereto, as machines other than those with reciprocating pistons may be suitable.

I claim:

1. In combination, a reversible valve fluid motor adapted for driving a vehicle, a prime mover, a fluid compressor driven by said prime mover, a high pressure connection between the exhaust port of said compressor and the intake port of said motor, a low pressure connection between the exhaust port of said motor and the intake port of said compressor, a high pressure reservoir in said high pressure connection, a low pressure reservoir in said low pressure connection, means controlling the operation of said compressor tending to maintain the pressure in said high pressure reservoir, substantially constant at a normal value, a relief valve connected to said high pressure reservoir, said relief valve being set to discharge at some pressure higher than said normal pressure and adapted thereupon to discharge fluid into said low pressure reservoir, means for reversing the valves on said motor to cause the motor to act as a compressor, a throttle in said high pressure connection between said high pressure reservoir and said motor, and valve means adapted when said throttle is closed to prevent the flow of fluid from said high pressure

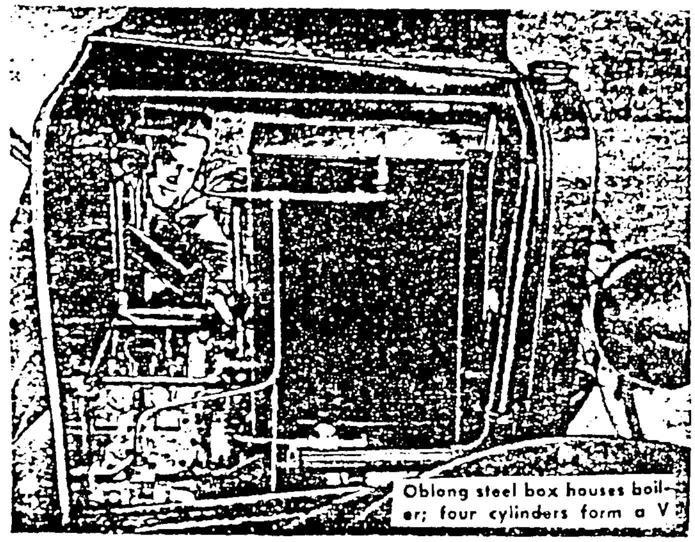
reservoir into said motor but to permit fluid compressed by the motor to be pumped back into the high pressure reservoir.

2. In combination, a vehicle, a valve controlled reversible fluid pressure motor for driving said vehicle, a prime mover, a compressor driven by said prime mover, a high pressure connection between the exhaust port of said compressor and the intake port of said motor, a low pressure connection between the exhaust port of said motor and the intake port of said compressor, a high pressure reservoir in said high pressure connection, a low pressure reservoir in said low pressure connection, means controlling the operation of said compressor tending to maintain the pressure in said high pressure reservoir substantially constant at a normal value, a relief valve connected to said high pressure reservoir, said relief valve being set to discharge at some pressure higher than said normal pressure and adapted thereupon to discharge fluid into said low pressure reservoir, means for reversing the valves on said motor to cause the motor to act as a compressor, a throttle in said high pressure connection between said high pressure reservoir and said motor, and a check valved connection shunting said throttle and acting independently of the setting of said throttle to permit fluid compressed by the motor to be pumped back into the high pressure reservoir.

ROBERT CADY BURT.

The CAR *that* RUNS *on* Air

FEBRUARY, 1945 POPULAR MECHANICS



Oblong steel box houses boiler; four cylinders form a V

POSTWAR motorists may climb into their automobiles, step on the air instead of the gas, and glide away swiftly and silently at 60 miles to the gallon with never a gear to shift or clutch to shove.

That's the prospect presented by Frank P. Perry, Los Angeles inventor, who has built a revolutionary car called a "Perrymobile."

Untried commercially, it's a novel combination of steamer and compressed air powered automobile. The motive power comes from a secret liquid, which vaporizes at a much lower temperature than water, and compressed air which serves as an ever-ready starting and reserve source of power.

Perry says he has driven several thousand miles with his machine which is mounted on an old Ford chassis. The Perrymobile weighs only 700 pounds—about 1,300 pounds less than standard automobiles powered by the conventional internal combustion engines. The engine installation alone weighs only 140 pounds.

The 30-horsepower four-cylinder engine is turned over by pressures instead of by the explosions that move the pistons of an ordinary auto engine. The Perrymobile engine is essentially the same as a steam engine. Inside each cylinder is a piston which moves up and down and is connected to the crankshaft.

The secret non-inflammable liquid which boils at about 150 degrees Fahrenheit, is heated by a burner which uses anything from butane gas to crude oil. The car will travel 60 miles at 30 miles an hour on one gallon of butane, the inventor claims. He says this fuel costs about 8½ cents a gallon. (He sets the top speed at "better than 70.") Only one quart of the secret liquid is required in the boiler because it is exhausted as vapor into the radiator, condensed and returned to the boiler.

The vapor passes through an intake valve into the top of the cylinder, and with a pressure of about 150 pounds per square inch pushes the piston down just as steam would do. At the bottom of the stroke the vapor exhausts through a port cut through the cylinder wall. As the piston starts up again a valve at the top of the cylinder lifts so the piston travels upward against no air pressure. At the top of the stroke more vapor is admitted which starts the piston down again.

Each of the four pistons supplies power to the rear wheels every time it is pushed down. This affords "two-cycle" operation instead of the conventional "four-cycle" shift in which each piston gets a shove only every second time it travels downward. Thus, Perry's four cylinders put out as many power impulses as an eight-cylinder auto engine.

The compressed air part of the power for the Perrymobile comes from a tank under the seat. That tank is kept full of compressed air supplied by a small air pump connected by a belt with the engine. The compressed air, which flows through the boiler into the cylinders, is used for a quick start and until sufficient vapor pressure has been built up to run the engine.

To operate this revolutionary automobile, you first open the fuel valve under the hood and the burners catch fire from a pilot light. Then you get in the car and pull down the throttle lever on the right side of the steering column. Compressed air from the tank flows into the cylinders and the car starts to move.

After you have driven a few blocks the flame in the boiler has built up sufficient vapor pressure so you can turn off the air. The pump quickly restores the pressure in the air tank. Heat in the boiler is regulated by an automatic valve.

Suppose you are driving out into the country and come to a long sloping downgrade. You close the throttle, just as you would in an ordinary car, and the flame goes low in the boiler, for no pressure is being used. Even while the engine is idling, air is pumped into the tank to maintain a constant pressure. To back up, a control on the left of the steering column changes position of the cams for reversing.

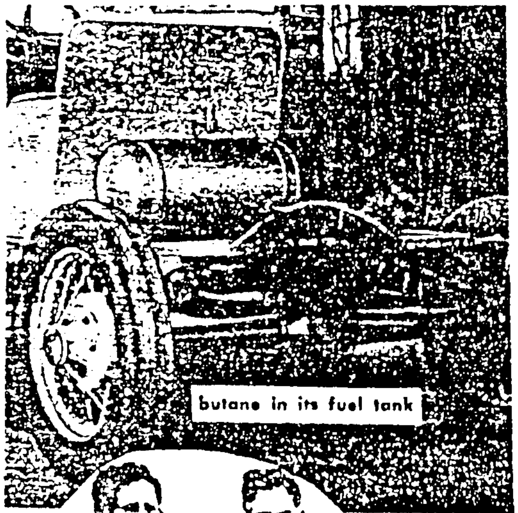
The inventor compares his single control for regulating speed and power to the control lever of an electric motor. The smooth operation of the Perrymobile is due to the fact that the power output is the same at all speeds—from 1 r.p.m. to 2,000. On a gasoline engine the horsepower is in direct ratio to the r.p.m.'s.

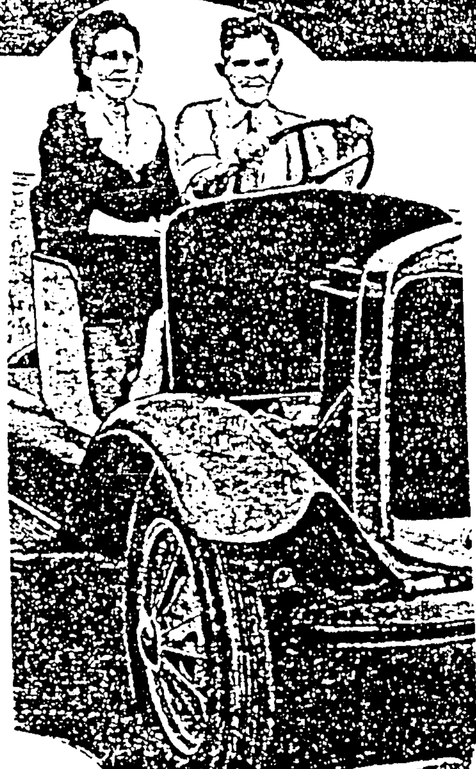
Even at full power with the throttle valve wide open, Perry says his engine runs so cool that the paint has never blistered on the cylinders. This makes the car adaptable to extremes of climate for the liquids in it do not freeze unless the mercury drops to 30 below zero. The engine turns over slowly. At 40 miles an hour it revolves only 800 times a minute compared with 2,000 or more for most present-day cars.

Other advantages of the Perrymobile, according to the inventor, are the "parts it does without." These include clutch, carburetor, spark plugs, distributor, coils, battery, fan, gear box and self starter. The car, of course, is equipped with brakes. Perry estimates that it will require less than one quart of lubricating oil a year. The Perrymobile makes no noise, smoke or smell. So smooth is its operation, he reports, that in a blindfold test it is impossible to tell when the car starts moving.

It cost Perry about $400 to build his

PERRY MOBILE

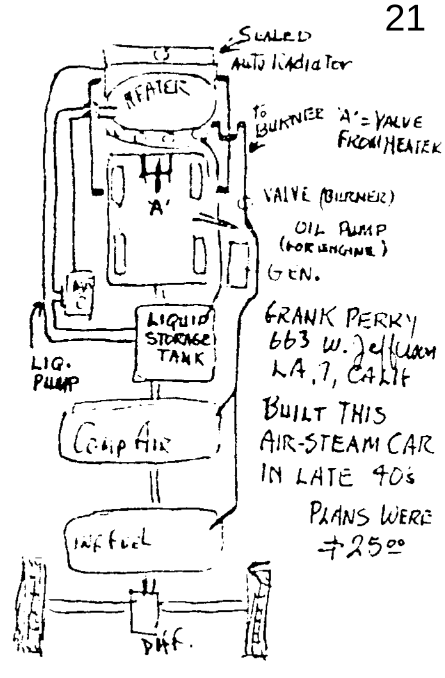




Built on an old Ford chassis,

lightweight automobile, but he says it should sell for much less if it gets into mass production—about $250. He believes the air-vapor engine can be used on helicopters and boats as well as automobiles.

The only floorboard pedal is the brake; a control on steering column regulates both speed and power



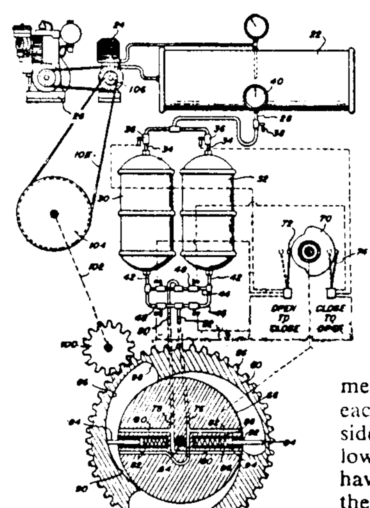
FRANK PERRY
663 W. Jefferson
LA. 7, CALIF
BUILT THIS
AIR-STEAM CAR
IN LATE 40's
PLANS WERE
$25.00

V-4 'L' HEAD · 30 HP
MODEL A FORD Rods & CRANK
2½" PISTON TIMING GEARS 3
DIFF. & REVERSE
AIR PUMP GAINS POWER FROM
DECELERATION
LIQ ST. TANK COLLECTS LIQ AS IT
CONDENSES IN RADIATOR
NO FURTHER DATA FEY 65

OFFICIAL GAZETTE

JUNE 21, 1960

2,941,608
POWER DEVICE
James E. Parrish, P.O. Box 154, Intercession City, Fla.
Filed Dec. 11, 1958, Ser. No. 779,621
3 Claims. (Cl. 180—6.2)



1. A power unit for a vehicle comprising a supporting frame, a pair of ground engaging wheels arranged in tandem on opposite sides of said supporting frame, wheel driving gears mounted in tandem with one on each of said wheels, a power driving gear supported in meshing engagement in between each pair of wheel driving gears on opposite sides of said frame, a single compressed air tank mounted on said frame for holding high pressure air, an air compressor to supply the initial high capital pressure to said compressed air tank connected with said compressed air tank, a prime mover connected with said air compressor, a pair of oil reservoirs on each side of said frame and supported thereby, one of each pair of reservoirs being initially filled with oil, the second of each pair being empty of oil when said first is full, a single discharge pipe connected at one end with said single compressed air tank, said discharge pipe having a throttle valve intermediate its ends and connected at its opposite end to cross-over lines connected to the upper portions of each of said pair of reservoirs, a fluid oil motor supported on each side of said frame, each fluid oil motor having a hollow casing rotatably mounted on its power shaft, one of said power driving gears integrally formed with each of said hollow casings, pipe means for conveying oil from the filled oil reservoir on each side of said frame to said fluid motor on the same side thereof, cross over pipe means for connecting the lower end portions of each of reservoir of each pair and having valve means associated therewith for controlling the flow of oil from said oil filled reservoir to the fluid motor and back to said empty reservoir, gear and pulley means connected with said power driving gears and drivingly connected to said air compressor by an automatic clutch connecting pulley for driving said compressor while the motors are running to keep the normal running air pressure in said compressed air tank, cam means connected with said rotating hollow casing for controlling said valves in the cross arms above and below said reservoirs whereby the flow of oil from one reservoir under high pressure to the other reservoir of each pair after passing through a fluid motor is maintained in unidirectional flow during alternating flow from one reservoir to the other of each pair.

United States Patent

[11] 3,589,126

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[21] Appl. No. 813,521
[22] Filed Apr. 4, 1969
[45] Patented June 29, 1971

[54] **POWER SYSTEM**
5 Claims, 1 Drawing Fig.

[52] U.S. Cl. .... 60/36, 60/59 T
[51] Int. Cl. F01k 25/00
[50] Field of Search .... 60/36, 59 T, 57 T

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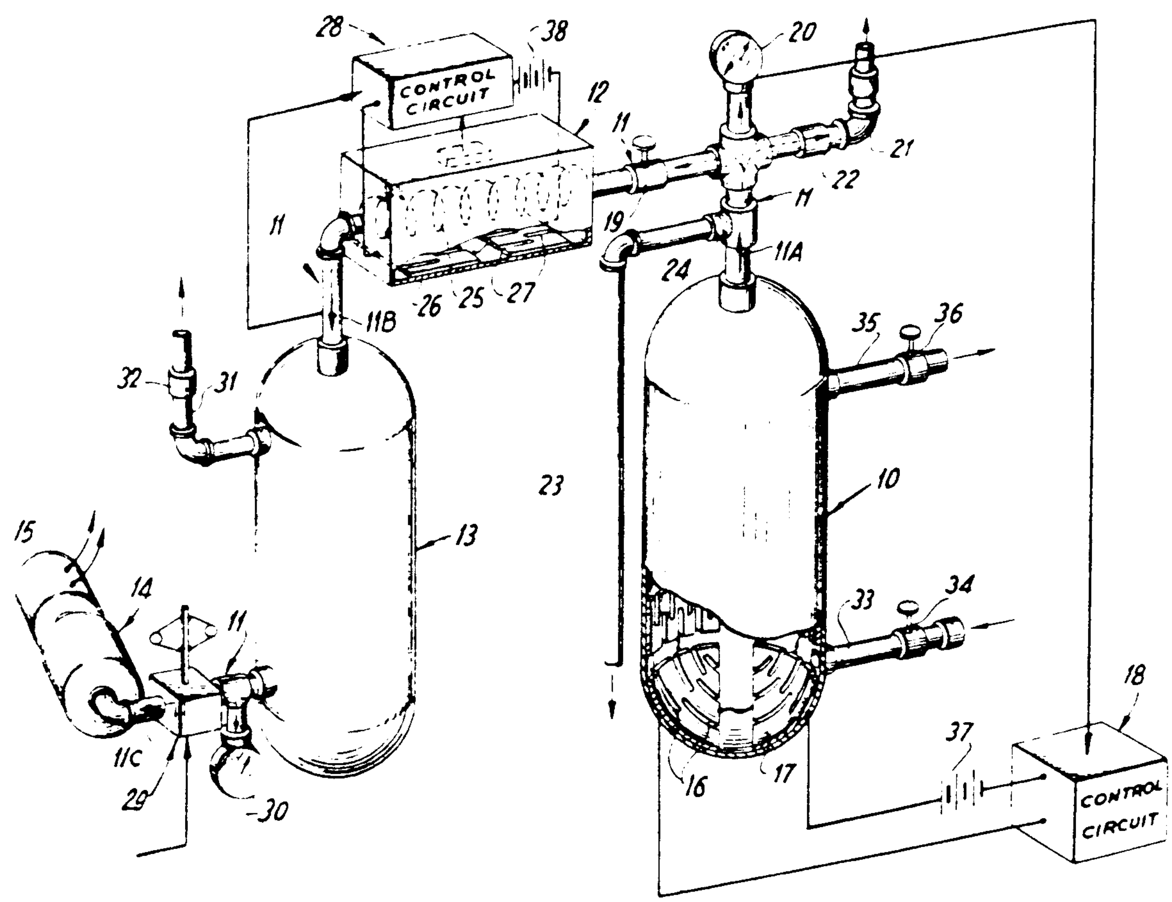
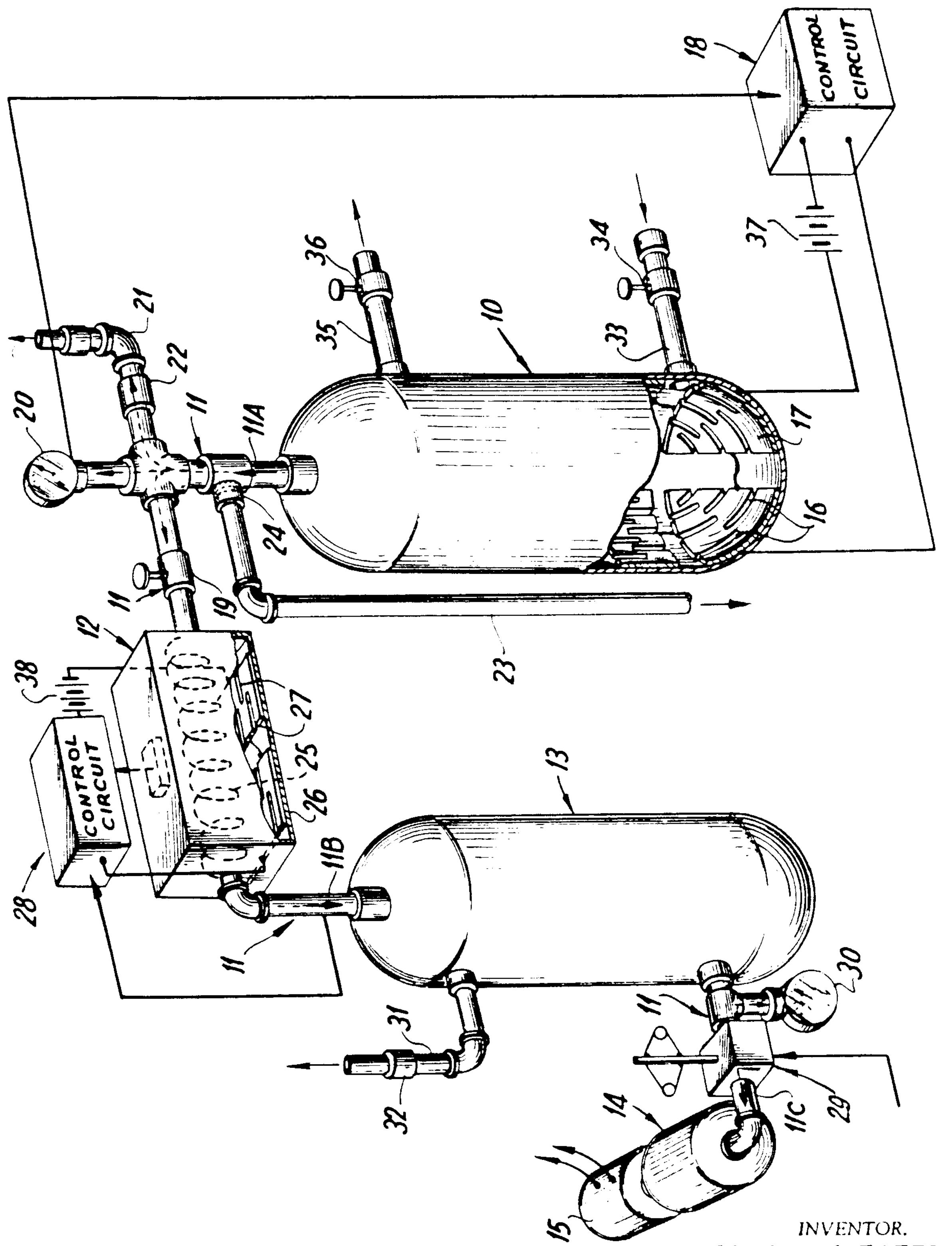
ABSTRACT: The production of power by the utilization of a supply of a gaseous working fluid under pressure to drive a turbine, wherein a substantially unlimited supply of working fluid is provided by the controlled evaporation of a reservoir of the working fluid in a liquefied state in accordance with pressure variations at an appropriate point in the system. Heating of the gaseous working fluid to a preselected temperature and pressure prior to utilization is advantageous.





INVENTOR.
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ATTORNEYS

POWER SYSTEM

BACKGROUND AND SUMMARY OF THE INVENTION

Ever since the inception of the industrial revolution, a constant search has been underway for new and improved ways and means for producing power. This is true even in today's highly industrialized and mechanized world, where the huge and varied demands for power have rendered the problem more acute than ever. One factor contributing to the widespread experimentation that is taking place today is the realization that the conventional fuels are likely to become exhausted in the not unforeseeable future at the ever-increasing rates at which they are being consumed. Of more immediate concern, however, and a major motivating factor towards increased experimentation is the danger to health and property resulting from the continuing uncontrolled pollution of our environment. Poisonous exhaust gases from the internal combustion engine are continually polluting the air we breathe. Waste products from petroleum processing plants are polluting our streams and rivers.

At a time when increasing pressures are being brought to bear on private industry by public and private interests to clean up the atmosphere, the automobile industry in particular has been hard pressed to come up with a practical solution to the problem. In order to meet certain safety standards set up for the industry, experimentation has resulted in the provision of special carburetors designed to eliminate or cut down on the contaminants which result from internal combustion. Obviously, these measures do not provide the desired solution to the problem and, moreover, they lead to less efficient combustion of the fuel with commensurate losses of power being the result.

The concept of an electrically driven car has received considerable attention. However, the major stumbling block to the application of this concept has been the industry's inability to provide a practical source of electrical energy in the quantities required. Storage batteries have been tried but represent a grossly inefficient source of energy.

In view of the shortcomings and limitations of the prior art, the instant invention has as an objective the provision of new and improved ways and means for producing power in a clean, safe and efficient manner.

It is another object of the instant invention to provide methods and instrumentalities for developing and channeling the power source potentialities of supercooled fluids such as liquid air.

It is more particular object of the instant invention to provide a practical solution to the problem of air pollution with particular reference to the effects of the internal combustion engine by the utilization of a media which will provide efficient and economical service without the danger of objectionable contaminus being introduced into the atmosphere.

Briefly and generally, the foregoing and other objects and advantages are accomplished in accordance with the instant invention by the provision of a system including a plenum chamber containing a gaseous working fluid in operative communication with a turbine and means for regulating the withdrawal of working fluid from the chamber to drive the turbine in accordance with demand for power, the pressure head in the plenum chamber being constantly maintained by the closely controlled conversion to its natural gaseous state of a reservoir of liquefied working fluid in accordance with pressure variations sensed at an appropriate point in the system. Advantageously, a heat exchanger is provided for raising the temperature of the workin[illegible] in its gaseous state to ambient or above prior to ent[illegible]um chamber, and a check valve is provided [illegible] the reservoir of liquefied working fluid and the heat exchanger to maintain one-way fluid flow towards the plenum chamber, pressure sensing for fluid conversion taking place on the upstream side of the check valve.

Having summarized the invention, a more detailed description follows with reference being had to the accompanying drawings which form a part of this specification and in which an exemplary embodiment of a power-producing system in accordance with the invention is diagrammatically illustrated.

DETAILED DESCRIPTION OF THE INVENTION

Turning now in detail to the accompanying drawings, the exemplary system illustrated therein comprises a tank **10** wherein the controlled evaporation of a reservoir of liquefied gas such as liquid air takes place, and from which the evaporated fluid is transported via high-pressure conduit system **11** through a heat exchanger **12** and thence to a plenum chamber **13** from which it is withdrawn, as needed, to drive a turbine **14** having an associated generator **15**. The invention will hereinafter be described in terms of a liquid air system for exemplary purposes only.

As mentioned above, the initial liquid air conversion takes place in tank **10**. It is necessary therefore that a supply of liquid air be stored and held ready for use within the tank. Accordingly, tank **10** is particularly adapted to maintain, as nearly as possible, the air in its liquefied state and is accordingly designed to prevent or retard heat transfer with the atmosphere. A Dewar vessel typifies the kind of storage tank presently contemplated. Of course, it is recognized that complete prevention of heat transfer may not be attainable and that some conversion of liquid to gas probably will occur due to uncontrollable heat transfer during normal periods of nonuse. The system is provided with safety pressure relief means described below for counteracting the possibility of an uncontrolled pressure buildup in the tank and in other parts of the system as well.

Tank **10** is provided with pressure-responsive heating means for heating the liquid air stored therein in order to effect the controlled evaporation of the liquid air in accordance with pressure variations in the system. As embodied herein, the aforesaid heating means comprises a plurality of electrical strip heaters **16** incorporated into the inner wall **17** of the tank **10** at and adjacent to its lower end. Control of the operation of the strip heaters is accomplished by means of a suitable control circuit diagrammatically illustrated at **18** which is adapted to energize and deenergize the strip heaters in response to instantaneous pressure conditions in the system. The control circuit means contemplated for use herein can comprise conventional instrumentalities such as pressure-sensing devices and switches arranged in a conventional manner. It is believed that the nature of the control circuit arrangement required to accomplish the desired heater control will be immediately apparent to one possessing ordinary skill in the art.

The tank **10** is coupled at its upper end to heat exchanger **12** by a first high-pressure conduit section **11A** including a check valve **19** for maintaining one-way flow through the system. The pressure-sensing elements of the control circuit **18** are illustratively inserted into the system at first conduit section **11A** on the upstream side of check valve **19**. Also coupled into the system at this point is a pressure gauge **20** for indicating the operating pressures in the tank.

In order to maintain safe pressure conditions in the tank, a pressure relief line **21** and valve **22** of the conventional type are operatively coupled into the system at conduit section **11A**. An additional safety factor is provided by the inclusion of a secondary pressure relief arrangement including a pressure relief line **23** and rupture disc **24** located at the juncture of line **23** and conduit section **11A**. Disc **24** is adapted to rupture when the pressure in the tank and hence in conduit section **11A** exceeds a preselected magnitude preferably greater than the operating pressure of pressure relief valve **22**. Thus, should pressure relief valve **22** malfunction or should its associated relief line **21** be incapable of providing the necessary pressure relief, the disc **24** will rupture providing a second escape route to the atmosphere via secondary pressure relief line **23**.

Heat exchanger 12 comprises in its illustrative form coils of high-pressure tubing 25 surrounded by a suitable heat transfer medium, e.g., water, which is encapsulated in an outer jacket or shell 26. Suitable controlled heater means is provided for maintaining a suitable temperature condition of the heat transfer medium. As embodied herein, the walls of the outer jacket 26 are provided with a plurality of electrical strip heaters 27 in like manner to the inner wall of tank 10.

In order to assure that the working fluid leaving the heat exchanger is at the desired temperature and pressure, control means 28 is provided for energizing and deenergizing heaters 27 to regulate heat transfer to the working fluid. The control means 28, which is diagrammatically illustrated, comprises means indicated by the arrow for monitoring the fluid in conduit section 11B as it departs the exchanger 12 and initiating the appropriate responses in a control circuit arrangement which would include suitable switching elements or equivalent means for providing the heater control to obtain the desired fluid conditions. The means for monitoring the fluid can include conventional fluid temperature, pressure and/or flow rate sensing devices (not shown) optimumly inserted within conduit section 11B. An alternative arrangement can find temperature-sensing devices within the heat transfer medium surrounding the exchanger tubes 25. The overall arrangement contemplated with its various components is believed to be well within the ordinary skill of the art and further description here is thought to be unnecessary. The temperature rise to be imparted to the air as it passes through exchanger 12 is a matter for determination through practice of the invention. At present, it is contemplated that the air will be b brought to a temperature at least coinciding with the ambient temperature of the atmosphere, it being understood, however, that the higher the temperature, the greater the pressures developed for utilization.

After it passes through heat exchanger 12, the heater air is channeled to the plenum chamber 13 via a second high-pressure conduit section 11B. The plenum chamber 13 functions as a form of reservoir to provide a ready supply of working fluid under pressure to drive air motor 14. No special requirements exist for the construction of the vessel serving as the plenum chamber other than may be dictated by pressure and temperature requirements. In other words, the wall of the vessel must be of a construction to withstand the internal pressures to which it will be subjected during operation, and, where the temperature of the working fluid is to be above ambient, the vessel should be of a type similar to the liquid air tank 10 in order to retard or limit heat transfer with the atmosphere. The optimum pressures and temperatures for the working fluid in plenum chamber 13 are matters for determination through practice of the invention. However, the nature of the turbine 14 and the power requirements of particular applications are obviously important factors to be considered in this regard.

Operatively connecting plenum chamber 13 with turbine 14 is the third or final high-pressure conduit section 11C. The pressurized working fluid of the plenum chamber is channeled, as needed, directly to the turbine via this conduit section. This is accomplished by means of a flow-regulating device such as the centripetal governor 29 situated in the conduit section 11C. In practice, this regulating device can be responsive to external control so that the operator will be able to directly regulate the power output of the system.

Situated in conduit section 11C just prior to the governor 29 in the illustrative embodiment is a pressure gauge 30 enabling the operator to maintain a close check on pressures at this critical point in the system.

As will be understood, in normal operation the rate of egress of working fluid from chamber 13 via conduit section 11C should equal the rate of ingress of fluid to the chamber via conduit section 11B. However, a safety factor is provided in the event that these rates do not coincide and excessive pressures tend to build up within the chamber. Pressure relief means including a pressure relief line 31 and conventional pressure relief valve 32 provides an auxiliary and immediate escape route to the atmosphere to counteract any such imbalance in the system.

Tank 10 is provided with means for the introduction of fresh supplies of liquid air when needed. An inlet line 33 having a shutoff valve 34 is provided at the lower end of the tank. Near the upper end of the tank is a vent line 35 having a similarly associated shutoff valve 36. When it is desired to introduce a fresh supply of liquid air into tank 10, the inlet line 33 is coupled to the supply source of the liquid air. Vent line shutoff valve 36 and inlet line shutoff valve 34 are then opened. The fresh supply of liquid air is then transported via inlet line 33 to the interior of the tank until it is filled to the desired level. As the tank is being filled, some of the liquid air will undoubtedly evaporate and escape through the vent line 35. However, as the inner wall 17 of tank 10 cools through contact with the supercooled liquid air, less evaporation will occur. When the desired level has been reached, the shutoff valves of the vent and inlet lines are closed and the system is ready for operation.

The manner in which the invention achieves its objectives will now be explained with the aid of the system illustrated herein. The operator of the system provides manual control over the centripetal governor 29 for the purpose of regulating the operation of the turbine. As the conduit section 11C is opened by governor 29, for greater power, greater volumes of working fluid are channeled to the turbine from the plenum chamber 13. The greater the rate of withdrawal of air from the plenum chamber, the greater the speed of the turbine and the greater the power output. Hence, direct control is provided over the power output of the system. The turbine, in turn, can be utilized to drive a generator 15 for the production of electrical energy for any purpose desired.

In order to obtain optimum efficiency and predictable response in the operation of the system, it is very important that the pressure within the plenum chamber be constantly maintained at the optimum operating condition determined for the particular application. As already discussed, pressure relief line 31 and valve 32 play an important role in preventing excessive pressure buildups within the chamber.

When working fluid is withdrawn from the plenum chamber 13 for operation of the turbine, a drop in pressure obviously occurs in the chamber. This pressure variation immediately results in fluid movement from heat exchanger 12 towards plenum chamber 13, the ultimate result being a pressure drop in liquid air tank 10 and conduit section 11A on the upstream side of check valve 19. This ultimate drop in pressure on the upstream side of valve 19 is picked up by the pressure-sensing devices of heater control means 18 actuating the appropriate mechanisms for energizing electrical strip heaters 16. Energization of strip heaters 16 results in the production of heat causing practically immediate evaporation of the supercooled liquid air in the tank back to its natural gaseous state. The gaseous working fluid thereby produced moves through the system towards the plenum chamber, passing through check valve 19 and heat exchanger 12 where its temperature is raised. When power is no longer required and the pressures in the system are again in equilibrium, the sensing devices of the heater control means 18 initiate the appropriate response to deenergize the strip heaters 16 and discontinue the evaporative process.

It should be apparent that individual control of the plurality of strip heaters by control means 18, or the utilization of variable heaters the rate of heat production of which can be closely controlled, may be a desirable feature of the invention. In normal operation a continuous withdrawal of working fluid from plenum chamber 13 will be taking place, but the rate of withdrawal will, in all probability, fluctuate. Individual heater control and/or the provision of controlled variable heaters will enable the rate of evaporation of liquid air to be more closely controlled with this in mind.

The evaporated working fluid after having undergone a change of state is still at a temperature far below that of ambient. By passing it through the heat exchanger 12, the

evaporated fluid can be raised to ambient temperature or above with concurrent increases in working pressures being developed to thereby greatly increase the operating efficiency of the system. As stated earlier, the optimum increase in temperature to be imparted to the working fluid can be determined through practice of the invention. Due to the increased pressures developed in the system on the downstream side of the check valve, lesser quantities of liquefied working fluid need be evaporated to meet power output requirements than would be the case if secondary heating were not included in the process. Accurate control of the rate of evaporation in terms of the need for actual working fluid is attained by sensing pressure variations on the upstream side of the valve.

In order to impart the desired temperature rise to the working fluid passing therethrough, the heat transfer medium, e.g., water, surrounding the high-pressure coils **25** of the exchanger **12** is maintained at a suitably high temperature. Maintenance of the appropriate temperature is accomplished automatically by the strip heaters **27** and control means **28**. The temperature, pressure and/or flow rate sensing elements of heater control means **28**, which are preferably suitably located in the conduit section **11B** leading from the heat exchanger **12** to the plenum chamber **13**, monitor the working fluid and initiate the appropriate response signal to either energize or deenergize one or more of the strip heaters **27** to attain the proper rate of heat transfer to achieve the desired temperature.

As a result of the rise in temperature of the evaporated air, greater pressures are developed for possible utilization in driving the turbine, these greater pressures being directly related to greater system efficiency. If it is desired to maintain fairly high temperatures above ambient within the plenum chamber **13**, the chamber should be constructed in a manner similar to liquid air tank **10**, i.e., to retard heat transfer with the atmosphere.

Both heater control means **18** and **28** can be powered by individual batteries as shown in the drawings at **37** and **38**, respectively, or they can be powered by electrical energy which has been shunted off the generator **15** for that purpose.

It will be seen from the foregoing that a relatively simple yet potentially highly efficient power system is provided. The liquid air takes up relatively little space compared to the tremendous working pressures which can be built up and utilized through proper channeling. Moreover, the exhaust of the system consists solely of air so that no harmful contaminants reach the atmosphere. Clean working energy is produced.

It is to be understood that the invention in its broader aspects is not limited to the specific elements, steps, techniques, combinations and arrangements herein shown and described, but departures may be made therefrom without departing from the scope and spirit of the invention as defined in the appended claims and without sacrificing its chief advantages. For example and without limitation, placement of the pressure sensing instrumentalities of the liquid air heater control means can be made at points in the system other than where illustrated herein, if such proves desirable. Thus, practice of the invention may determine that in certain instances it may be desirable to sense pressure variations directly within the plenum chamber. As another example, different types and forms of heat exchangers can be utilized in the practice of the invention as long as they are suitable for providing the desired temperature rise to the working fluid. Moreover, the invention is not to be limited to the particular working fluid discussed herein, i.e., air, provided, however, that a proper substitute is available. Other variations and departures from the disclosure herein within the scope of the invention will undoubtedly occur to those skilled in the art through practice of the invention.

What I claim is:

1. A system for producing power comprising
a. a turbine having a fluid outlet;
b. a plenum chamber filled with a working fluid in its natural gaseous state at a pressure greater than the pressure at said turbine outlet,
c. first fluid conduit means operatively connecting said plenum chamber and said turbine, said conduit means including flow-regulating means responsive to external control for regulating the mass rate of flow of working fluid to said turbine;
d. an insulated fluid vessel containing a reservoir of said working fluid in a liquefied state, said vessel having internal heater means for heating said liquefied working fluid;
e. heater control means including pressure sensing means for sensing fluid pressure conditions at a suitable point in the system and automatically selectively energizing and deenergizing said heater means in accordance with said conditions; and
f. second fluid conduit means operatively connecting said fluid vessel and said plenum chamber such that reductions in fluid pressure within said plenum chamber result in the energization of heater means, the gasification of liquefied working fluid within said insulated vessel, and the movement of said gasefied working fluid towards said plenum chamber.

2. A system in accordance with claim **1** further comprising heat exchanger means operatively coupled into said second fluid conduit means for raising the temperature of said gasified working fluid as it moves from said insulated vessel to said plenum chamber.

3. A system in accordance with claim **2**, said second fluid conduit means comprising check valve means intermediate said insulated vessel and said heat exchanger means operative to prevent fluid movement towards said insulated vessel.

4. A system in accordance with claim **3**, said pressure-sensing means of said heater control means being operatively positioned in said system to detect pressure variations on the upstream side of said check valve means.

5. A system in accordance with claim **4**, further comprising control means for monitoring said working fluid as it departs said exchanger means and regulating the rate of heat transfer to said working fluid to obtain the desired temperature and pressure conditions for said fluid.

United States Patent

Maruyama

[15] **3,704,760**

[45] **Dec. 5, 1972**

[54] **ELECTROPNEUMATIC PROPELLING SYSTEM FOR VEHICLES**

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[73] Assignee: **Oscar Kogyo Kabushiki Kaisha**, Tokyo-To, Japan

[22] Filed: **June 22, 1971**

[21] Appl. No.: **155,422**

[52] **U.S. Cl.** .... **180/66 B**, 180/65 A
[51] **Int. Cl.** **B60l 11/14**
[58] **Field of Search** .... 180/65 A, 66 B, 65 R

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Primary Examiner—Gerald M. Furlenza
Assistant Examiner—R. B. Johnson
Attorney—Holman & Stern

[57] **ABSTRACT**

A system for electropneumatically propelling a vehicle comprises essentially a pneumatic motor and an electric motor. The pneumatic motor is driven by the air supplied under pressure from a source of compressed air such as a cylinder or a liquid-air tank, while the electric motor is driven by a storage battery in which is stored electrical energy produced by a generator, which is driven by a turbine operated by air supplied under pressure from the source of compressed air. Both the pneumatic motor and the electric motor are connected to transmission means linked to the driving wheels of the vehicle. The transmission means includes first and second differential means adapted for controlling the supply of air to the pneumatic motor and for causing the pneumatic motor to rotate in the same direction as a preselected rotational direction of the electric motor.

6 Claims, 11 Drawing Figures

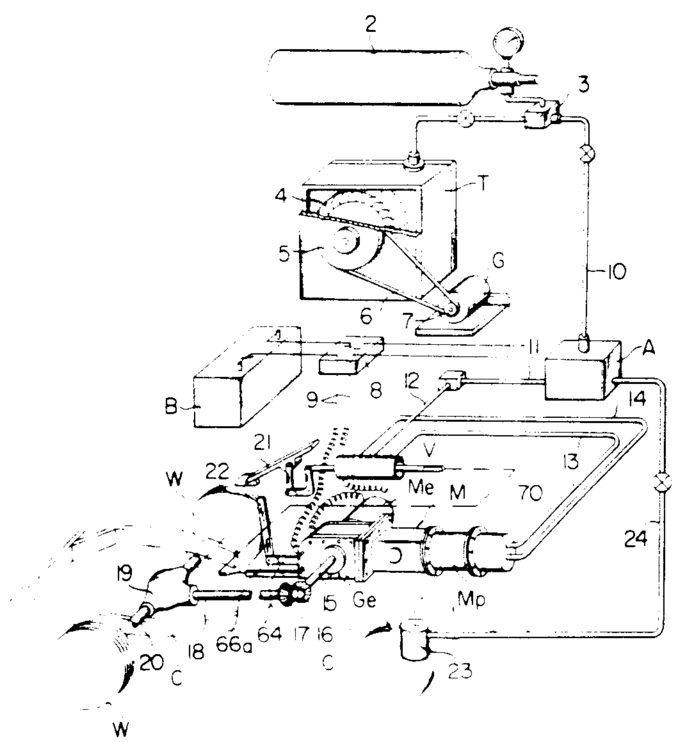

FIG. 1

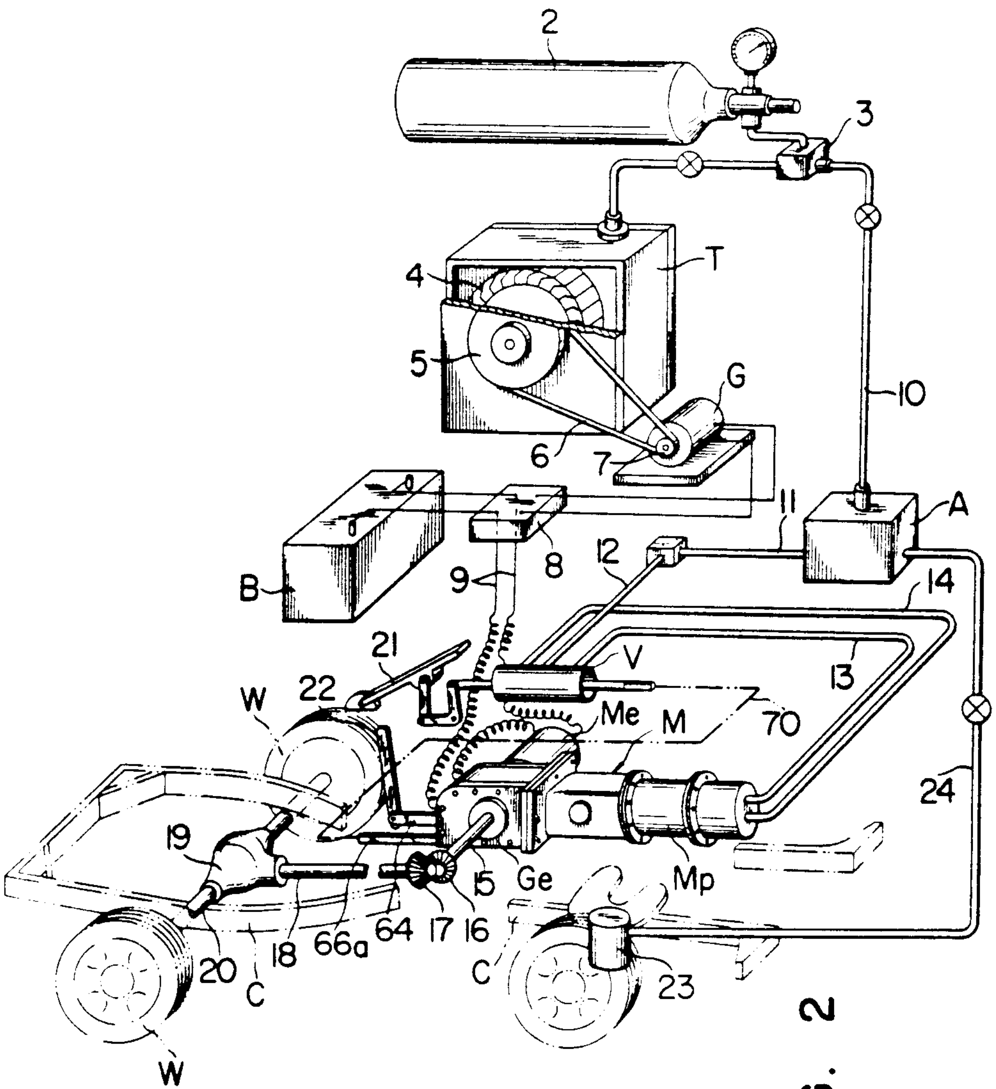
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FIG. 2

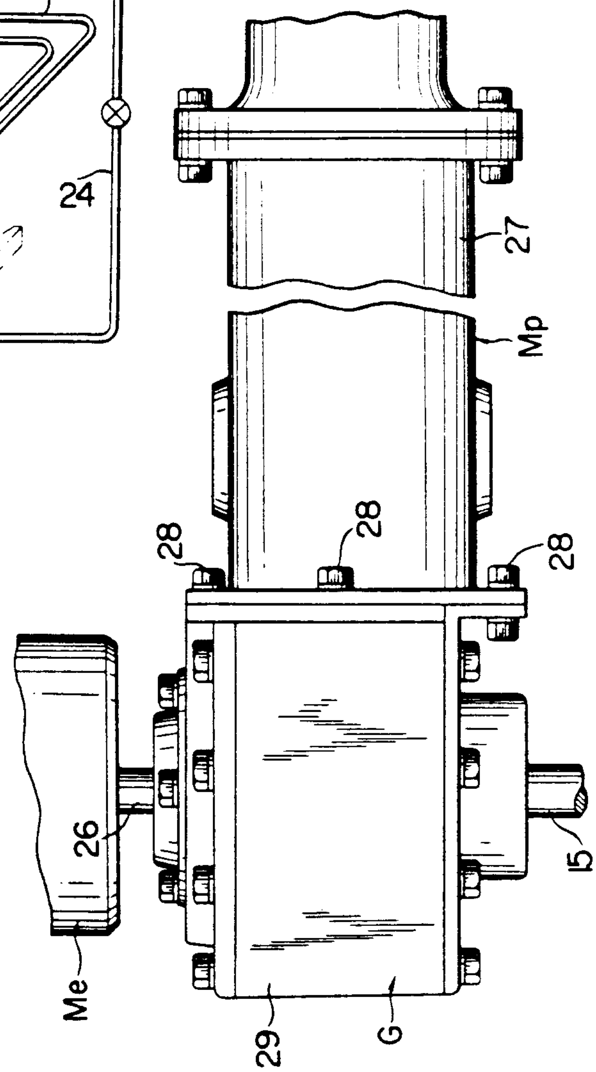
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FIG. 3

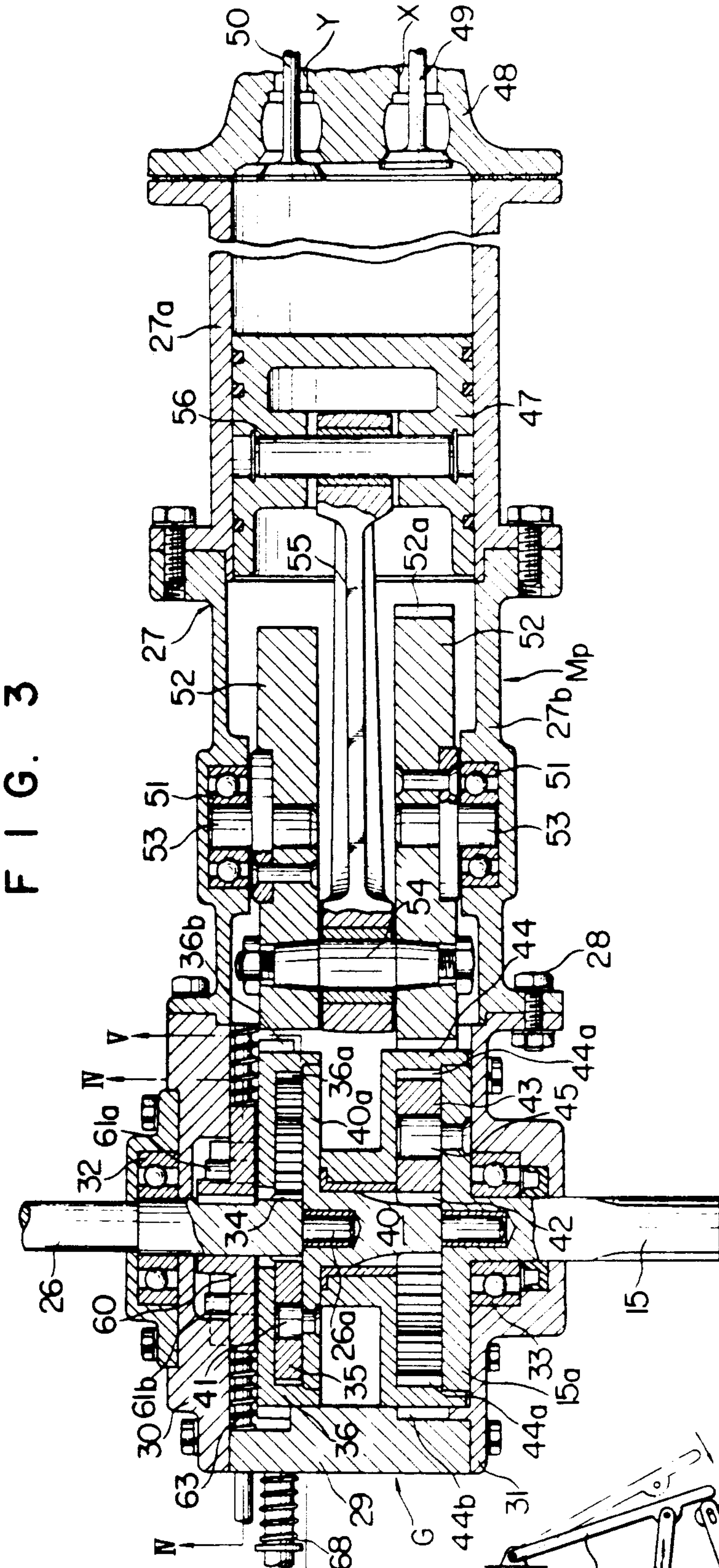


FIG. 4

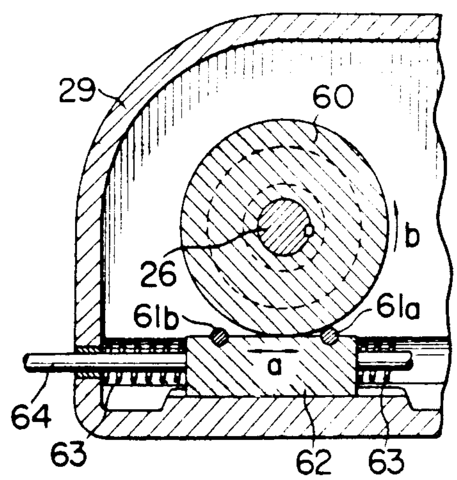


FIG. 8

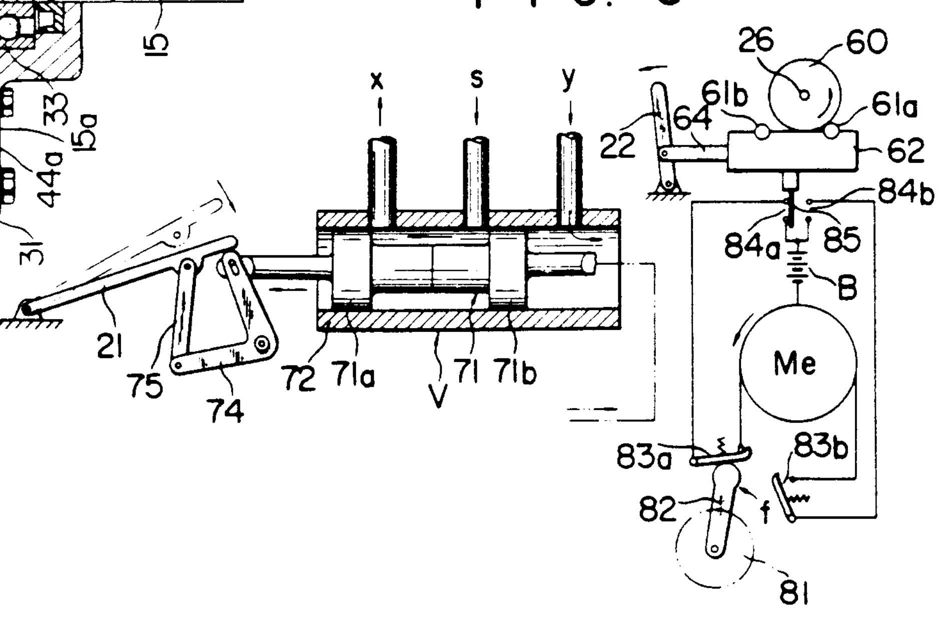
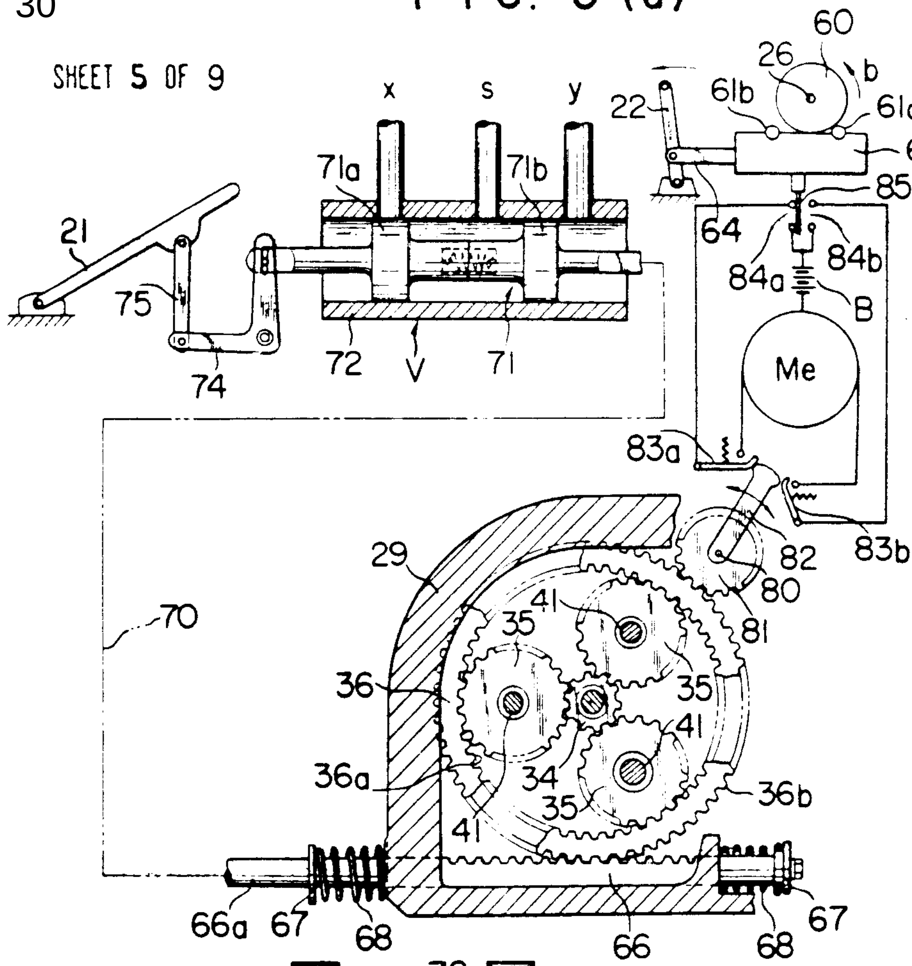


FIG. 5(a)



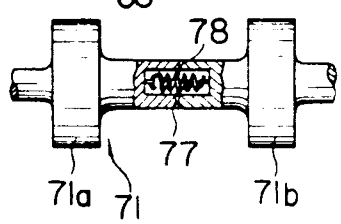
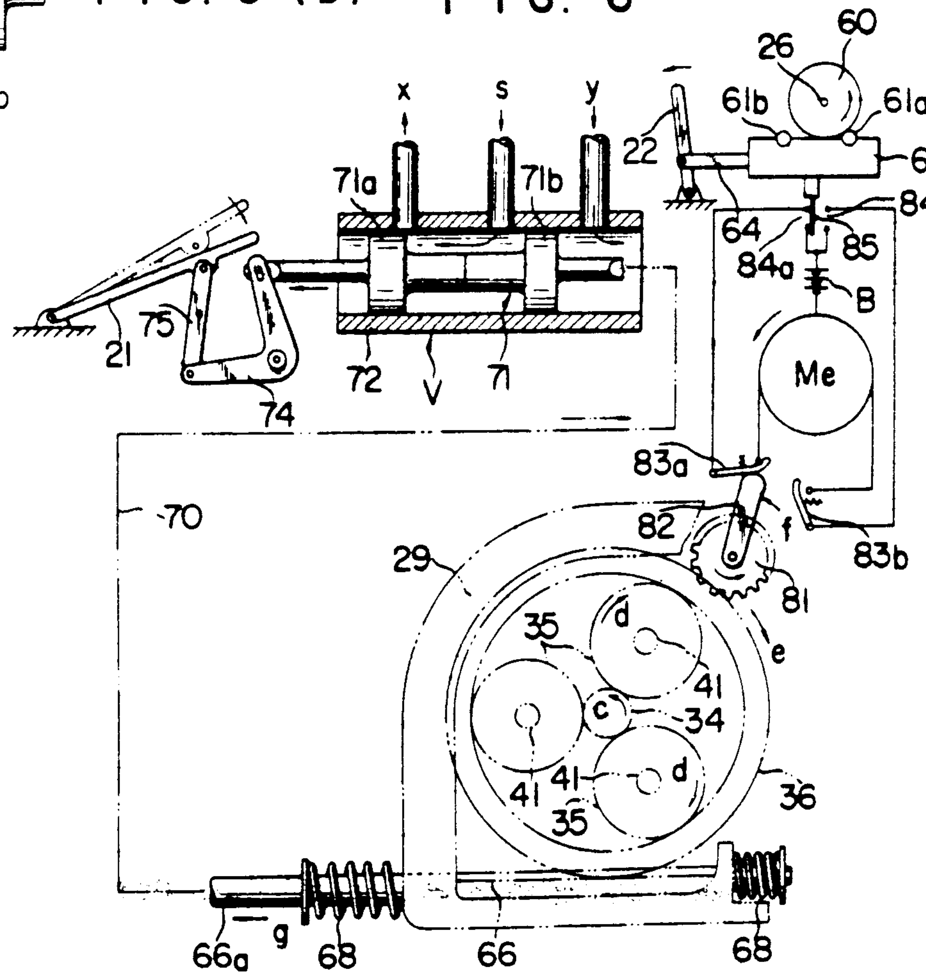


FIG. 5(b)

FIG. 6



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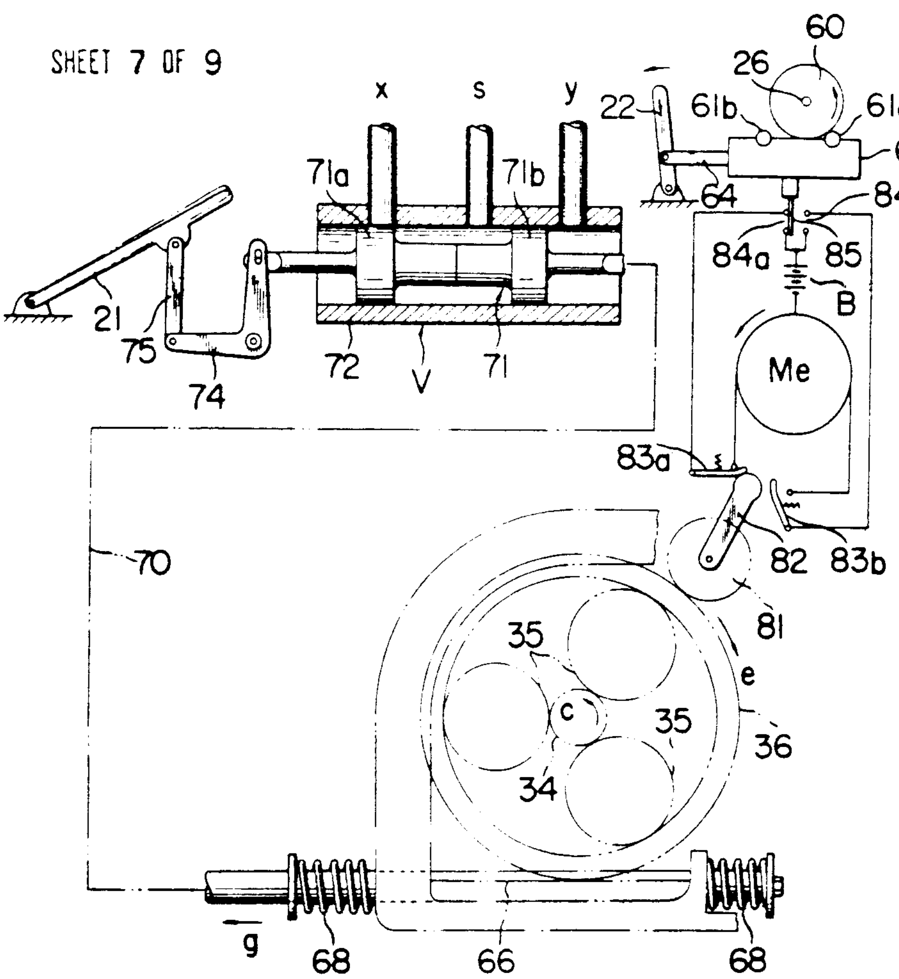

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F I G. 9

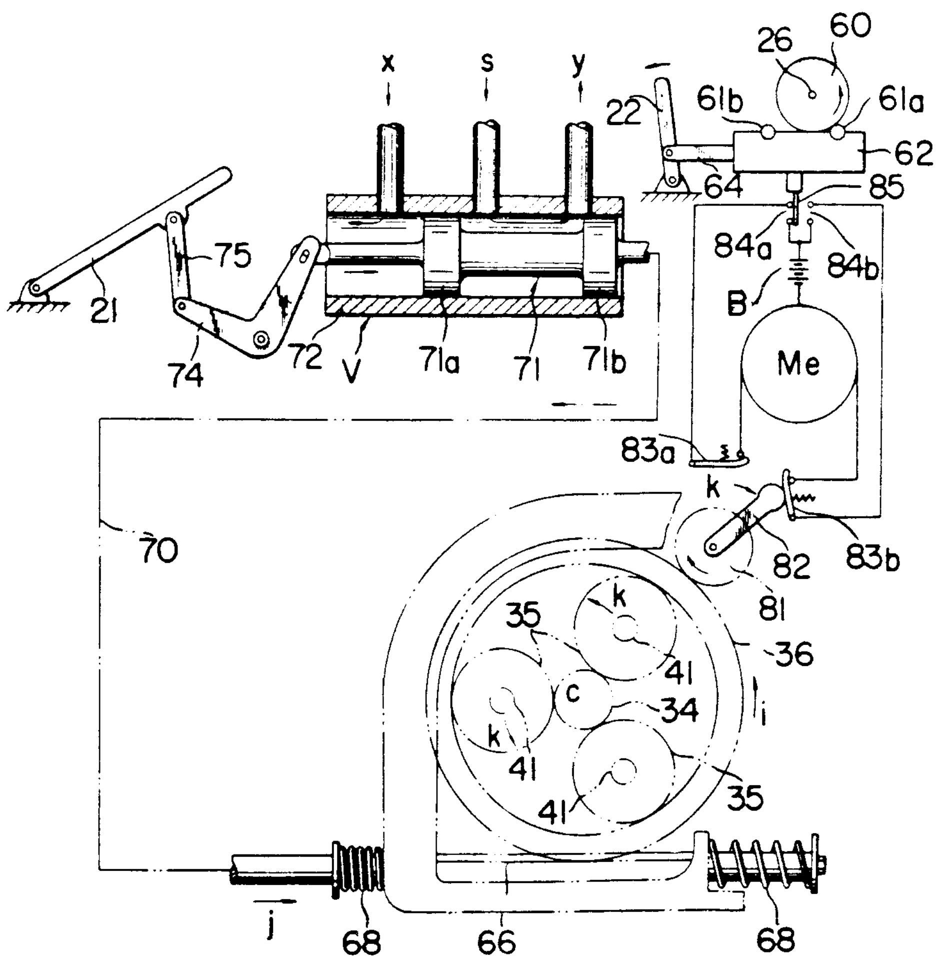

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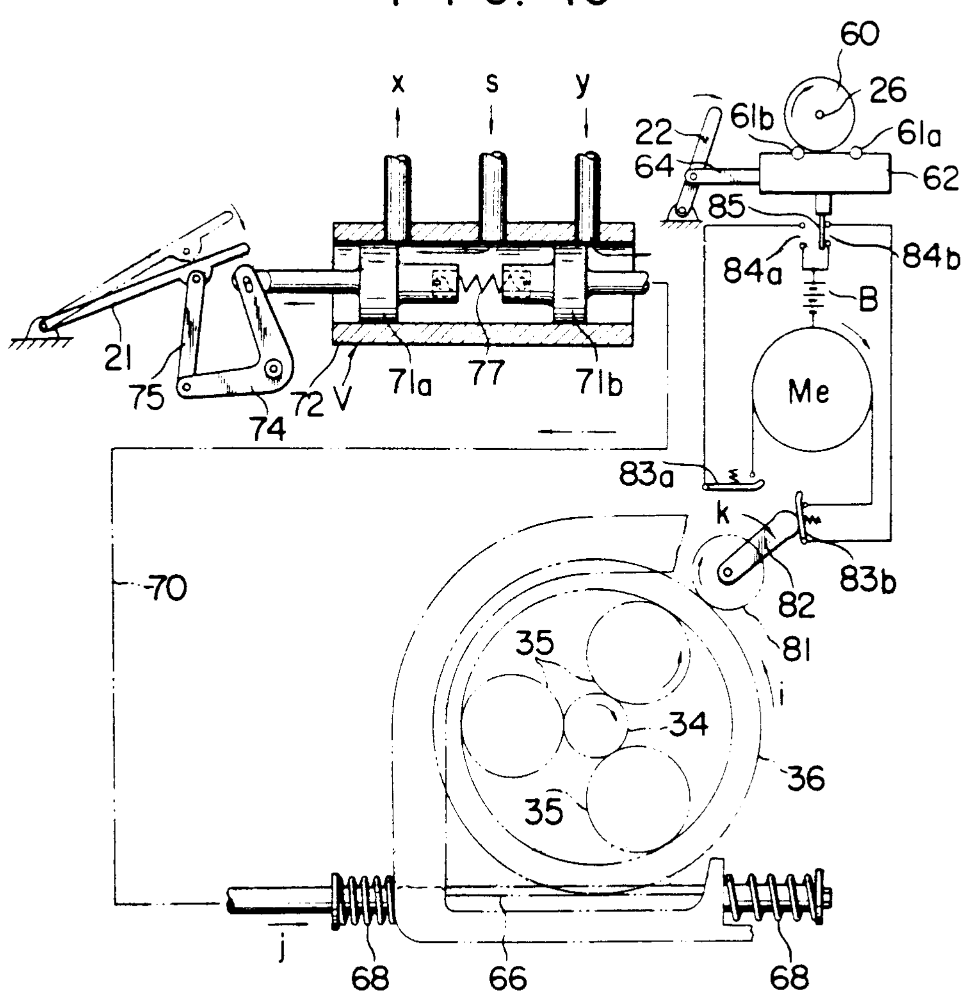
FIG. 10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ELECTROPNEUMATIC PROPELLING SYSTEM FOR VEHICLES

BACKGROUND OF THE INVENTION

The present invention relates to a novel system for electropneumatically propelling a vehicle.

Automakers now are making desperate efforts to solve the air pollution problem caused by motor vehicles that are powered by conventional internal combustion engines. The need is an urgent one in the United States and other heavily motorized countries of the world where more and more rigid governmental pollution regulations are being imposed upon the motor vehicles.

While the search may continue for new, nonpolluting power sources, much attention is now being focused upon electric cars, various types of which have been proposed, and some already have been manufactured on a considerable scale. A major factor hampering the growth of the electric cars, however, is that their batteries are too expensive and do not have enough capacity to drive the vehicles for any practical purposes. Nevertheless, the fact remains that electric cars are one clear solution to the air pollution problem, being admirably suitable for some transportation services.

SUMMARY OF THE INVENTION

It is accordingly an object of the present invention to provide a novel electropneumatic propelling system for vehicles to be run with any likelihood of polluting the air.

Another object of the invention is to provide an electropneumatic propelling system for vehicles wherein an electric motor and a pneumatic motor are provided in place of a conventional engine or its equivalent, the power delivered by the electric motor and that by the pneumatic motor being suitably combined for propelling the vehicle.

Another object of the invention is to provide an electropneumatic propelling system for vehicles wherein one and the same source of compressed air is utilized to drive both the pneumatic motor and the electric motor, the latter being driven via a pneumatic turbine, generator and storage battery.

Still another object of the invention is to provide an electropneumatic propelling system for vehicles wherein the electric motor and pneumatic motor are both coupled to transmission means of such organization that, upon depression of an accelerator pedal linked to a changeover valve means, the pneumatic motor is rotated only in the same direction as a preselected rotational direction of the electric motor.

A further object of the invention is to provide an electropneumatic propelling system for vehicles wherein the transmission means is associated with the changeover valve means in such a manner that both the electric motor and the pneumatic motor are driven simultaneously only when the delivery of large torque is required and that the vehicle is automatically propelled by the electric motor alone when it attains a cruising condition.

A further object of the invention is to provide an electropneumatic propelling system for vehicles wherein the electric motor and the pneumatic motor are so adapted that, when the vehicle goes downhill, the former automatically stops rotation while the latter

serves as a substantial "engine brake" to arrest the accelerated motion of the vehicle.

An additional object of the invention is to provide an electropneumatic propelling system for vehicles so designed that it requires an electric motor of comparatively small output and a storage battery of comparatively small capacity.

Other objects and advantages will appear from the following detailed description taken in connection with the appended drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings:

FIG. 1 is a schematic perspective view showing the layout of an electropneumatic propelling system in accordance with the present invention;

FIG. 2 is a fragmentary plan view showing the manner in which an electric motor and a pneumatic motor are coupled to transmission gear means in the electropneumatic propelling system of FIG. 1;

FIG 3 is a longitudinal sectional view showing the inner details of the transmission gear means and the pneumatic motor of FIG. 2;

FIG. 4 is a sectional view taken along the plane of line IV—IV in FIG. 3;

FIG. 5(*a*) is a schematic view indicating a condition of some pertinent mechanisms of the electropneumatic propelling system prior to the start-up of the vehicle, the differential means in the drawing being taken along the plane of line V—V in FIG. 3;

FIG. 5(*b*) is an enlarged detail view, partly in section, of a valve spool of FIG. 5;

FIG. 6 is a view similar to FIG. 5 indicating a start-up condition of the vehicle;

FIG. 7 is also a view similar to FIG. 5 indicating a cruising condition of the vehicle;

FIG. 8 is a view similar to FIG. 5 (except for the absence of the differential means, etc.) indicating a condition when the vehicle is going uphill;

FIG. 9 is a view similar to FIG. 5 indicating a condition when the vehicle is going downhill; and

FIG. 10 is also a view similar to FIG. 5 indicating a condition when the vehicle is running in reverse.

DETAILED DESCRIPTION OF THE INVENTION

Referring now to the drawings, and in particular to FIG. 1 thereof for a description of a preferred embodiment of the invention illustrated therein by way of example, a cylinder or vessel 2 filled with liquefied air supplies air under pressure to a turbine T via a pressure regulator 3 and so forth, thereby to rotate a rotor 4. The rotation of this turbine rotor 4 is conveyed to a generator G via suitable transmission means (5, 6 and 7, for example). The electrical energy thus produced by the generator G is collected in a storage battery B via voltage regulating means 8.

The reference character M generally indicates a prime mover (in a broader sense of the term) for the vehicle in accordance with the present invention, comprising an electric motor *Me* and a pneumatic motor *Mp*. The electric motor *Me* is rotatable in both directions driven by the electrical energy supplied from the storage battery B via conductors 9, as hereinafter described in detail. The pneumatic *Mp* is driven by the air that is supplied under pressure from the aforesaid

cylinder 2 and flowing through a conduit 10 (which may have a valve), accumulator A, conduit system 11 and 12, changeover valve V, and conduits 13 and 14. This pneumatic motor Mp is of a reciprocating piston type, as described later in greater detail.

Both the electric motor Me and the pneumatic motor Mp are coupled to transmission gear means Ge, whose output shaft 15 is connected via a pair of bevel gears 16 and 17 to an input shaft 18 of differential gear means 19 on an axle 20 of driving wheels W of the vehicle. The wheels W are thus driven by the combined powers of the electric motor Me and the pneumatic motor Mp delivered from the output shaft 15. These motors and the transmission gear means may be suitably mounted upon a vehicle chassis C. An accelerator pedal 21 is suitably linked to the changeover valve V, while a forward reverse control lever 22 is coupled to the transmission gear means Ge. A shock-absorber 23, suitably installed between the vehicle body and wheels, may be of a piston and cylinder type, the cylinder being filled with air so that pressure will build up therewithin due to the vertical oscillations of the body during running of the vehicle. This pneumatic pressure is directed to the accumulator A via a conduit 24.

As illustrated in greater detail in FIG. 2, the transmission gear means Ge is connected to the electric motor Me by means of an input shaft 26 of the former. The pneumatic motor Mp has a cylindrical housing 27 which extends at right angles to the input shaft 26 and output shaft 15 of the transmission gear means Ge, the housing 27 being secured to its casing or gearbox 29 by adequate means 28.

As shown in the sectional view of FIG. 3, both ends of the casing 29 of the transmission gear means Ge are closed by covers 30 and 31. The cover 30 rotatably supports the input shaft 26 by means of a bearing 32, while the cover 31 rotatably supports the output shaft 15 by means of a bearing 33. At the inner end of the input shaft 26 there is formed a sun wheel 34 that is a part of first planet differential means hereinafter to be described.

As illustrated in both FIGS. 3 and 5, the sun wheel 34 is in mesh with a plurality of planet wheels 35 which are themselves meshed with the internal teeth 36a of a ring gear 36. An intermediate shaft 40 is provided coaxially inside of the input shaft 26. This intermediate shaft 40 is integrally formed with a disk 40a on which the aforesaid planet wheels 35 are rotatably supported by shafts 41, and which loosely receives a pin 26a projecting from the inner end of the input shaft 26. At the other end of the intermediate shaft 40 there is formed another sun wheel 42 constituting a part of second planet differential means provided for the transmission gear means Ge. This sun wheel 42 is meshed with planet wheels 43, while these planet wheels are in mesh with the internal teeth 44a of a ring gear 44. Shafts 45 supporting respective planet wheels 43 are imbedded at their ends in a disk 15a that is integral with the output shaft 15. The aforesaid ring gears 36 and 44 also have external teeth 36b and 44b, respectively.

Also as shown in FIG. 3, the pneumatic motor Mp has a piston 47 slidably installed in the cylinder 27a which is a part of the overall pneumatic motor housing 27. A pair of valve means 49 and 50 are provided in a cylinder head 48 for the control of air supplied to and discharged from the pneumatic motor Mp. These valve means will open and close their respective air ports X and Y by means of known cam mechanisms or the like. Adjacent the cylinder 27a, there is provided a crankcase 27b which also forms a part of the overall pneumatic motor housing 27, and which, as above mentioned, is secured to the transmission gear casing 29 by the means 28.

A pair of bearings 51 within the crankcase 27b rotatably support the shafts 53 of parallel disks 52, one of the disks having teeth 52a on its periphery which are in mesh with the external teeth 44b of the ring gear 44. Eccentrically and imbeddedly fixed to both disks 52, a crankpin 54 is linked to a pin 56 of the piston 47 by means of a connecting rod 55. As this piston 47 is caused to reciprocate by the supply of air, the disks 52 are both rotated by the crankpin 54 via the connecting rod 55. This rotation of the disks 52 is imparted to the ring gear 44 via the intermeshing teeth 52a and 44b, and thence to the output shaft 15 via the planet wheels 43. The same rotation is also conveyed to the input shaft 26 via the intermediate shaft 40 and the first planet differential means.

Inasmuch as both the electric motor Me and the pneumatic motor Mp may be driven simultaneously, it is imperative that they rotate the output shaft 15 in the same direction. Accordingly, in order to prevent the pneumatic motor Mp from rotating in the direction opposite to a preselected rotational direction of the electric motor Me, there is provided the following device according to the present invention.

As illustrated in both FIGS. 3 and 4, this device is broadly comprised of a disk 60, which is keyed or otherwise suitably mounted on the input shaft 26, and a carriage 62 supporting a pair of rollers 61a and 61b to be moved into contact with the rim of the disk 60. Slidable in the direction of the arrow *a* as indicated in FIG. 4, the carriage 62 is normally held at a neutral position by means such as helical compression springs 63. A rod 64 extends from one end of this carriage 62 and, projecting out of the transmission gear casing 29, is coupled to the aforementioned forward reverse control lever 22 as illustrated in FIG. 1. In the condition of FIG. 4, in which the carriage 62 is displaced leftward as viewed in the drawing by the operation of the control lever 22, the input shaft 26 is permitted to turn only in the direction of the arrow *b*, which is assumed to be forward, because of the roller 61a wedged in between the carriage 62 and the disk 60. By turning the control lever 22 in the opposite direction, the carriage 62 is displaced rightward as viewed in FIG. 4 so that its roller 61b is caught between the carriage and the disk 60. The input shaft 62 in this instance is capable of making only a reverse rotation.

With reference now made to FIG. 5(*a*) in particular, the external teeth 36b of the ring gear 36 of the first planet differential means are meshed with a rack 66 which is slidable on the inner surface of the casing 29 and which is normally kept at its centered or neutral position by such means as helical compression springs 68 installed between stops 67 and casing 29. An extension 66a of this rack 66 is connected to a spool 71 of the aforesaid changeover valve V via a system of linkages 70 (not fully illustrated)as in FIGS. 1 and 5(a). The casing 72 of the changeover valve V is in the shape

of an open ended cylinder, in which the spool **71** is slidably installed. This valve casing **72** is formed with a port S leading from the source of compressed air via the conduit **12** and so forth, a port x in communication with the intake port X of the pneumatic motor Mp via the conduit **14**, and a port y in communication with the exhaust port Y of the same via the conduit **13** (refer to FIG. 1). The opposite end of the spool **71** is coupled to the accelerator pedal **21** via a bell crank **74** and link **75**.

As illustrated in greater detail in FIG. 5(b), the spool **71** is made up of two separable portions **71**a and **71**b that are pulled together by means of a tension spring **77**. Thus the opposing ends **78** of these portions are ordinarily closely contacting each other.

Referring back to FIG. 5(a), the external teeth **36**b of the ring gear **36** of the first planet differential means are also in mesh with a pinion **81** on a shaft **80** suitably supported by the casing **29**. This pinion **81** is fixedly provided with an arm **82** which is swingable to operate a pair of normally open electrical switches **83a** and **83b**. The switch **83a** is closed when the arm **82** makes a counterclockwise turn as viewed in FIG. 5(a), while the switch **83b** is to be closed when the arm turns clockwise. The switch **83a** is included in a circuit adapted for the forward rotation of the electric motor Me, which comprises the motor Me, battery B and contact pair **84a**. The other switch **83**b is included in a circuit for the reverse rotation of the same, comprising the motor Me, battery B and contact pair **84b**. A movable contact **85**, which is made to move in coordination with the motion of the mentioned carriage **62**, closes the contact pair **84a** upon leftward movement of that carriage **62**, and the contact pair **84b** upon rightward movement of the same as viewed in FIG. 5(a). The movable contact **85** is not necessarily secured to the carriage **62** as in the drawing, but may be secured to the rod **64** or even directly to the control lever **22**, although in this latter case the relative positions of the contact pairs **84a** and **84b** may have to be modified accordingly.

FIG. 5(a) illustrates a condition prior to the start-up of the vehicle, in which the spool **71** of the changeover valve V is at its central or neutral position thus discommunicating the ports x and y from the port S. The rack **66** also is centered. The arm **82** of the pinion **81** keeps both of the electrical switches **83a** and **83**b opened, and the control lever **22** is turned leftward as viewed in FIG. 5 thereby to permit, upon closure of the switch **83a**, the forward rotation of the electric motor Me. By the leftward displacement of the carriage **62**, the roller **61**a is caught between the carriage and the disk **60** so as to permit only the forward rotation (indicated by the arrow b) of the shaft **26**. The movable contact **85** keeps the contact pair **84a** closed.

FIG. 6 is illustrated to show a start-up condition, in which the accelerator pedal **21** is depressed as indicated by the arrow in the drawing, thereby causing the leftward displacement of the spool portion **71a** of the changeover valve V. At this instant the ports S and x are intercommunicated so that the compressed air from the port S is made to flow into the intake port X of the pneumatic motor Mp via the conduit **14** (refer to FIGS. 1 and 3). The resultant reciprocation of the piston **47** and hence the rotation of the disks **52** is transmitted to the output shaft **15** via the ring gear **44** and the planet wheels **43**. The same rotation imparted to the ring gear **44** is also conveyed to the input shaft **26** via the intermediate shaft **40**. Since, however, the input shaft **26** then is permitted to rotate only in the forward direction as above mentioned, the disks **52** of the pneumatic motor Mp at the start of operation are turned in the direction that will cause the forward rotation of the output shaft **15**.

Upon forward rotation of the output and input shafts **15** and **26**, the sun wheel **34** of the first planet differential means starts rotating in the direction indicated by the arrow c in FIG. **6**, thereby causing the rotations of the planet wheels **35** in the direction of the arrow d. Should the ring gear **36** be immovable, the planet wheels **35** would revolve along the internal teeth of the ring gear **36** while rotating about their own shafts **41**. But the ring gear **36** is only retained by the springs **68** through the rack **66**, so that the great load applied at the time of vehicle starting will at first cause no substantial displacement of the planet wheels **35** relative to the ring gear **36** but will rotate the latter in the direction of the arrow e via the former. By this turn of the ring gear **36** and hence of the pinion **81** the arm **82** is swung in the direction of the arrow f thereby closing the electrical switch **83a**. As a result the electric motor Me is set in operation almost at the same time with the start-up of the pneumatic motor Mp to provide a great overall starting torque to the output shaft **15**.

The above explained initial turn of the ring gear **36** also causes the rack **66** to move leftward in FIG. **6** (indicated by the arrow g) against the right hand spring **68**. This leftward movement of the rack **66** stops when the force exerted thereto by the ring gear **36** is counterbalanced by that of the spring **68**. Thereupon the planet wheels **35** start revolving around the sun wheel **34** to transmit the rotation of the input shaft **26** to the output shaft **15**, while this output shaft **15** is also rotated by the pneumatic motor Mp, so that the vehicle now starts up. The mentioned leftward displacement of the rack **66** causes the portion **71b** of the valve spool **71** to move leftward as viewed in FIG. **6** via the system of linkages **70**.

All these actions described in the preceding three paragraphs take place almost in an instant. Since the vehicle in the above instance of start-up is under no greater load than that applied when it goes uphill, the port x of the changeover valve V is not fully opened by the spool **71** as illustrated in FIG. **6**. The pneumatic motor Mp does not therefore operate at its full capacity. Following the start-up the vehicle will run under gradually decreasing load, with the result that both the rack **66** and the spool **71** move rightward in FIG. **6** to provide a cruising condition illustrated in FIG. **7**.

Referring now to FIG. **7**, in which the rack **66** is still slightly displaced leftward by the clockwise turning force of the ring gear **36** (indicated by the arrow e), the arm **82** of the pinion **81** keeps the switch **83a** closed so that the electric motor Me is in motion. In order to ensure the closure of the switch **83a** whether the arm **82** is swung greatly as in FIG. **6** or only slightly as in FIG. **7**, it is necessary, or at least desirable, that the movable contact of the switch **83a** extend along the course of swing of the leading end of the arm and that this leading end be so formed as to make some elastic deformation.

During this cruising condition the rack **66** is only slightly moved in the direction of the arrow *g* as already mentioned, so that the valve spool **71** also is moved correspondingly leftward as in FIG. 7, the accelerator pedal **71** being slightly depressed. The valve spool **71** then closes the port *x* to stop the supply of air to the pneumatic motor M*p*, with its output decreased with the gradual closure of the port. Thereafter the vehicle will be propelled solely by means of the electric motor M*E*.

Illustrated in FIG. **8** is a condition when the vehicle is going uphill, or when the vehicle runs under greater load than that at the time of start-up (refer to FIG. 6). The valve spool **71** is this case moves farther leftward than its position illustrated in FIG. 6, thus substantially fully opening the port *x*. The rack **66** also travels toward the leftward extremity of its permitted stroke thereby to correspondingly urge the spool portion **71***b* in the arrow marked direction in FIG. 8.

FIG. **9** shows a condition when the vehicle is going down-hill. In this case the planet wheels **35** are compulsorily revolved counterclockwise (indicated by the arrows *h*) due to the naturally accelerated rotation of the vehicle wheels that is transmitted back through the output shaft **15**. The accompanying counterclockwise rotation of the ring gear **36** (indicated by the arrow *i* in FIG. **9**) causes the rack **66** to move in the direction of the arrow *j* and the arm **82** to swing in the direction of the arrow *k* via the pinion **81**. The switch **83***b* is now closed and the switch **83***a* opened, whereupon the forward rotation of the electric motor M*e* terminates. The closure of the switch **83***b* does not initiate the reversed rotation of the motor Me since the Contact pair **84***b* is kept closed. Because of the great turning force conveyed from the wheels of the vehicle going downhill, the rack **66** is considerably strongly urged in the direction *j* thereby causing a proportionately great displacement of the spool **71** of the changeover valve V, so that its ports S and *y* are now intercommunicated. Since, on the other hand, the piston **47** of the pneumatic motor M*p* is made to compress the air trapped within the cylinder **27***a* by the rotation of the output shaft **15** and so forth, the pneumatic motor operates substantially as a so-called "engine brake" to reduce the accelerated motion of the vehicle.

A condition showing the reverse movement of the vehicle is illustrated in FIG. **10**, in which the forward reverse control lever **22** is turned rightward as viewed in the drawing thereby to cause the roller **61***b* upon the carriage **62** to be wedged in between the carriage and the disk **60**. As a result the input shaft **26** is permitted to turn only in the reverse direction as indicated by the arrow in the drawing, while the contact pair **84***b* is closed by the movable contact **85**.

Upon depression of the accelerator pedal **21**, the portion **71***a* of the valve spool **71** moves leftward (as viewed in FIG. **10**) off the other spool portion **71***b* so that the ports S and *x* are intercommunicated. The air is now fed under pressure into the pneumatic motor M*p* from its intake port X, thereby setting the same in operation. Since the input shaft **26** is then permitted to move only in the reverse direction, the disks **52** of the pneumatic motor M*p* also are turned in the corresponding direction. This rotation of the disks **52** is conveyed to the sun wheel **34** via the intermediate shaft **40** and thence to the ring gear **36** via the planet wheels **35**, the ring gear **36** then being turned counterclockwise (as indicated by the arrow *i* in FIG. **10**) virtually for the same reasons as those set forth already in connection with FIG. 6. The arm **82** of the pinion swings as indicated by the arrow *k* thereby closing the switch **83**b and hence initiating the rotation of the electric motor M*e* in the reverse direction.

The rightward movement of the rack **66** (indicated by the arrow *j*) is conveyed through the system of linkages **70** to the valve spool portion **71***b*, which then is moved oppositely to the other spool portion **71***a* thereby expanding the spring **77**. The rack **66** in this case is not displaced to such an extent as that attained in the event of the downhill drive of the vehicle explained with reference to FIG. **9**, so that the spool portion **71***b* does not quite close the port *y*. Hence the air supplied to the pneumatic motor M*p* is properly exhausted therefrom. With both the electric motor M*e* and the pneumatic motor M*p* thus rotated in the reverse direction, the vehicle will be driven in reverse by the large torque delivered by the output shaft **15**.

I claim:

1. An electropneumatic propelling system for a vehicle comprising a source of compressed air, a turbine driven by air supplied under pressure from said source, a generator driven by said turbine, a storage battery adapted for storing electrical energy produced by said generator, an electric motor driven by electrical energy supplied from said storage battery, a pneumatic motor driven by air supplied under pressure from said source, valve means adapted for the control of air flow from said source to said pneumatic motor, and transmission means to which are connected both said electric motor and said pneumatic motor, and which is coupled to the driving wheels of the vehicle.

2. An electropneumatic propelling system according to claim **1**, wherein said transmission means comprises an input shaft coupled to said electric motor, an output shaft coupled to the driving wheels of the vehicle, an intermediate shaft between said input shaft and said output shaft, first planet differential means at one end of said input shaft associated with said valve means and with said electric motor, and second planet differential means at one end of said output shaft to which is conveyed the rotation of said pneumatic motor.

3. An electropneumatic propelling system according to claim **2**, wherein said first planet differential means comprises a ring gear loosely mounted on said input shaft, a sun wheel fixedly mounted on said input shaft, and a plurality of planet wheels meshing with both said ring gear and said sun wheel and respectively mounted on shafts fixed to an integral disk of said intermediate shaft, said ring gear being also in mesh with a rack adapted for actuation of said valve means.

4. An electropneumatic propelling system according to claim **3**, wherein said rack is coupled to one of two separable portions of a spool member of said valve means, the other portion thereof being linked to an accelerator pedal of the vehicle, and said two separable portions being interconnected by means of an elastic member.

5. An electropneumatic propelling system according to claim **2**, including means for causing said pneumatic motor to rotate in the same direction as a preselected

rotational direction of said electric motor, said means comprising a disk fixedly mounted on said input shaft of said transmission means, a displaceable carriage adjacent the rim of said disk, a pair of rollers fixedly mounted on said carriage with a spacing therebetween, a manually actuable control lever linked to said carriage thereby to cause either one of said rollers to be wedged in between said carriage and said disk and hence to permit said input shaft to rotate only in the preselected direction, a movable contact actuated in coordination with the movement of said control lever to close the corresponding one of contact pairs respectively inserted in circuits for forward and backward rotations of said electric motor, said circuits respectively having electrical switches the corresponding one of which is closed by a member associated with said first planet differential means upon rotation in the preselected direction of said input shaft by the power supplied by said pneumatic motor.

6. An electropneumatic propelling system according to claim 1, including pneumatic shock-absorbing installed between the vehicle body and wheels, and an accumulator connected between said source of compressed air and said valve means, said buffer means being communicated with said accumulator in order to supply pneumatic pressure produced therein to the latter.

* * * * *

Automotive Engineering, November 1976

Compressed-Air Hybrid

Some of your readers, who studied the interesting article on Electric Vehicles in the August edition, may like to spare a passing thought on the merits of pneumatic vehicle propulsion. Almost all the arguments and relationships presented in the paper still hold good if compressed air is substituted for electricity. But the real interest lies in the differences.

It is not likely that a reasonable range can be achieved on stored compressed air alone, so that a hybrid, with an engine covering mean power demand to drive the air compressor, needs to be considered.

The potential transmission efficiency of a pneumatic transmission is high compared to the electric analogue and if heat from the charging engine exhaust is used to heat the driving air, a transmission efficiency in excess of unity is possible. In fact, with an open circuit layout, such an arrangement is necessary to avoid freezing in the air motor exhaust.

There is no limit to the charge or discharge rates and no difficulty in applying 100% regenerative braking. What is more, it is a lot simpler to store air at, say, 100 psi than to store electric energy in a high speed flywheel. There is also no battery charging loss and no battery weight; the storage weight of compressed air is small and it all but disappears if chassis members are made to double as air receivers. The weight of air motors is infinitesimal compared to electric drive motors and nothing cleverer than a few valves is needed for propulsion control. There would be two or more air motors, so there is no need for a differential either.

The charging engine, which either stops or runs at constant speed and load, can be tuned to maximum efficiency and minimum exhaust emission and if the economy steps found necessary for the electric vehicles are applied to the pneumatic hybrid, it should be possible to attain quite phenomenal fuel mileages.

Finally, the traditional skills of the automotive industry could readily encompass the development and production of such a vehicle.

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MECHANICAL ENGINEERING / MARCH 1983

Air Compressor-Driven Autos

To the Editor:

It's a mystery why energy conservation efforts have focused on a dead phoenix such as the electric car, and the friction-plagued flywheel, while our factories use a readily storable 95 percent pollution-free energy called compressed air.

Coupling a compressor to the existing engine of an auto or truck would provide compressed air to air motors driving the vehicle wheels. The savings in energy would result from shutting the engine off in an off/on cycle whenever the pressure in a small air storage vessel was high enough to move the vehicle from stopped position such as a stop-light, and as air demand decreases in downhill/slowing for a stop condition. With the proper air circuit design, dynamic braking would be achieved with the air motors in a compressor mode for the latter conditions.

Since the air supply could be varied to meet the road needs, the vehicle engine would operate at a constant, most economical rpm to replenish the air. I guesstimate a 10-minute supply of compressed air in the storage vessel would be able to allow the engine to restart on demand as the vehicle is moving to replenish the supply. Naturally, air motor engine starting is desirable.

An existing auto could be modified with off-the-shelf compressor, air motors, valves, etc. The only design required is for adapters to attach the above components, and a special foot-operable pressure regulator which would replace the accelerator pedal.

With this arrangement, no transmission is necessary.

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April 23, 1968 C. A. MANGANARO **3,379,008**

FLUID PRESSURE SYSTEM FOR OPERATING A VEHICLE DRIVE

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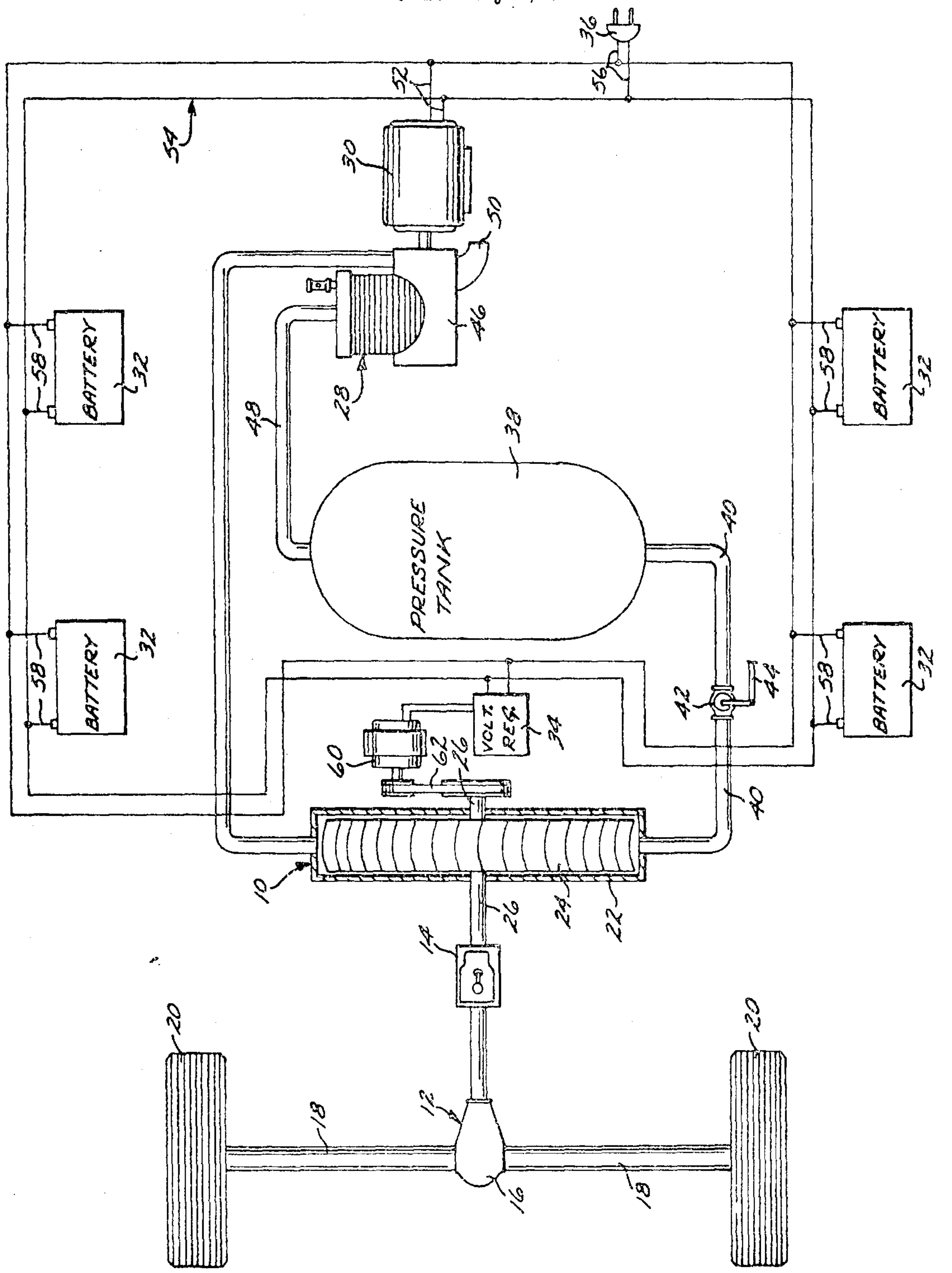

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FLUID PRESSURE SYSTEM FOR OPERATING A VEHICLE DRIVE

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Filed May 5, 1966, Ser. No. 547,838
4 Claims. (Cl. 60—57)

ABSTRACT OF THE DISCLOSURE

A turbine drive system for propulsion of a vehicle and in which the turbine is driven by air compressed by a compressor which is driven by an electric motor, the motor being operable by batteries in circuit with an alternator which is driven by the turbine.

The present invention relates to a fluid pressure system for operating a vehicle drive, and more particularly to a system in which a battery operated motor is utilized to operate the means for supplying fluid under pressure to operate the vehicle drive.

It is an object of the present invention to provide a fluid pressure system for operating a vehicle drive to propel a vehicle, and which is economical to operate, relatively inexpensive to manufacture, and completely free of the noxious and irritating gases which characterize the operation of conventional internal combustion engines.

Another object of the invention is to provide a fluid pressure system of the aforementioned character which is characterized by simplicity of construction, relatively few moving parts, and a simple to operate fluid flow throttle for regulating the speed of the vehicle.

Yet another object of the invention is to provide a fluid pressure system of the aforementioned character which includes a compressor for supplying compressed air to a reservoir coupled to a turbine which is connected to the vehicle drive, and which further includes a battery operable electric motor to drive the air compressor. The battery circuit includes means for detachably coupling the circuit to a separate source of enregy for periodically recharging the batteries, so that the vehicle can be driven during the day under battery power, the circuit plugged into a separate source of electrical energy for overnight recharging of the batteries, and the vehicle again driven the following day on the recharged batteries.

Other objects and features of the invention will become apparent from consideration of the following description taken in connection with the accompanying drawings, in which:

The lone drawing figure is a diagrammatic view of a fluid pressure system for operating a vehicle drive according to the present invention.

Referring now to the drawing, there is illustrated a fluid pressure system 10 for operating a vehicle drive 12. The driven vehicle (not shown) may be a boat, aircraft, motor vehicle or the like, and for illustration is sometimes referred to herein as an automobile.

The vehicle drive 12 includes a transmission 14, a differential 16 connected to and operated by the transmission 14, axle sections 18 rotatable by the differential 16, and wheels 20 carried by the axle sections 18 for rotation to propel the associated vehicle or automobile. The wheels 20 are usually the rear wheels of the automobile, the front wheels being omitted from the drawings for brevity.

The differential 16 is conventional in charcter, being operative to permit the wheels 20 to rotate at different speeds, as when the automobile is rounding a corner. The transmission 14 is operative to select a desired vehicle

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speed or to back the automobile, as is well known. Since these components are so well known in the art, details of their construction and operation are omitted for brevity, it being important primarily to note that they are merely exemplary of one form of vehicle drive which is operable by the fluid pressure system 10.

The fluid pressure system 10 includes, generally, a turbine 22 having a vaned rotor 24 whose shaft 26 is connected to the transmission 14, the rotor 24 being responsive to fluid under pressure to operate the vehicle drive 12, as will be seen. The system 10 further includes a means or apparatus 28 which is operative to supply fluid under pressure to the turbine 22; an electric motor 30 for operating the apparatus 28; a plurality of electric batteries 32 in circuit with the motor 30 for energization thereof; generating means connected to the rotor shaft 26 for generating electrical energy for the batteries 32; a voltage regulator 34; and an electrical plug 36 or the like for detachably coupling the battery and motor circuit to a separate source of energy for periodically recharging the batteries 32, as will be described in more detail below.

The turbine 22 is exemplary of one form of air motor which is operative to convert pressurized fluid into the mechanical energy necessary to rotate the drive or rotor shaft 26 for propelling the automobile. The turbine 22 is only diagrammatically shown since the details of its construction are conventional. The turbine rotor 24 is driven at relatively high rotational speeds by pressurized fluid passing from a pressure tank, receiver, or reservoir 38 through a conduit 40 and under the control of a throttle valve 42 located in the conduit 40.

The valve 42 is of conventional construction, including a movable section (not shown) which is operative by a throttle linkage 44 to close off and thereby regulate the flow of fluid through the conduit 40. The linkage 44 extends into a position for easy operation by the vehicle driver so that the speed of the vehicle can be closely controlled.

The fluid preferably utilized in the fluid pressure system 10 is compressed air which is compressed by means of a compressor 46, the compressor 46, the reservoir 38, and the valve 42 comprising the previously mentioned apparatus 28 for supplying compressed air to the turbine 22. The compressor 46 is preferably a conventional positive displacement piston type, and is connected to the reservoir 38 by a conduit 48. In addition, if the vehicle with which the system 10 is associated operates at comparatively high speeds, the intake end of the compressor 46 is also provided with an air intake scoop 50 to receive ram air for compression.

The compressor 46 is mechanically coupled to the electric motor 30 which, as previously indicated, is energized by electrical energy from the batteries 32. The motor 30 is conventional in construction and is connected by suitable electrical leads 52 to a motor-battery circuit 54, the plug 36 being connected to the circuit 54 by leads 56, and the batteries 32 being connected to the circuit 54 by leads 58.

When the driven vehicle is not being operated, the circuit 54 can be utilized for recharging the batteries 32. This is done by connecting the plug 36 to an outside or independent source of electrical energy (not shown), such as the usual household circuit of a residence. Since the system 10 illustrated is a direct current system, the usual alternating household current would have to be suitably rectified to operate the motor 30 and recharge the batteries 32. However, the system 10 could alternatively be made an alternating current systems, if desired, except for rectification of the charging current to the batteries 32, as will be apparent. Normally the motor 30 is op-

erated by electrical energy drawn from the batteries **32** through the electrical circuit **54**.

A number of other conventional components are also used in conjunction with the reservoir **38**, motor **30** and batteries **32**, but their description has been omitted for brevity and because the need for their inclusion will be apparent to those skilled in the art. For example, a suitable switch (not shown) is incorporated in the circuit **54** in order to connect and disconnect the motor **30** from the batteries **32**; safety fuses (not shown) would be incorporated in the circuits where needed; and pressure relief valves would be used with the compressor **46** and reservoir **38** to vent dangerously high pressures.

A voltage regulator **34** is incorporated in the circuit **54**, being electrically coupled to a generator **60** which is operative in the manner of the usual automobile generator to recharge the batteries **32** under certain conditions. The generator **60** is coupled by a chain or pulley drive **62** to the rotor shaft **26** so that during operation of the turbine **22**, the generator **60** applies electrical energy to the batteries **32** at such times as the batteries are not discharging at a high rate, as during a downhill run of the vehicle. Only limited recharging of the batteries **32** is possible during operation of the vehicle, the batteries **32** being periodically recharged by coupling of the circuit **54** to the separate source of electrical energy, as above-indicated.

Although not shown, the reservoir **38** can also be provided with means enabling its connection to a separate source of compressed air, such as would be available in an automobile service station. This would permit the reservoir **38** to be pressurized to the desired level without running the motor **30** and compressor **46**, as on initial start-up. Ordinarily however, the pressure level in the tank **38** can be maintained for a considerable period of time so that such outside pressurization will usually not be necessary.

In operation, the circuit **54** is coupled to the vehicle owner's residential electrical circuit for overnight charging of the batteries **32**, the capacity and number of batteries **32** preferably being such that the vehicle can be operated at moderate speeds during the day without additional charging. Upon disconnection of the circuit **54** from the household circuit, a suitable switch (not shown) is operated to connect the motor **30** to the batteries **32** to thereby operate the compressor **46**. When the appropriate pressure level is reached in the reservoir **38**, the operator actuates the throttle linkage **44** to open the throttle valve **42** and thereby supply compressed air to the turbine rotor **24**. Consequent rotation of the rotor **24** drives the wheels **20** to propel the vehicle, as will be apparent.

From the foregoing it is seen that a fluid pressure system has been provided for operating a vehicle drive, and which for this purpose utilizes a turbine operated by fluid

under pressure provided by an apparatus **28** which is driven by an electric motor **30** deriving its energy from a plurality of batteries **32**.

Various modifications and changes may be made with regard to the foregoing detailed description without departing from the spirit of the invention or the scope of the following claims.

I claim:

1. A fluid pressure system for operating a vehicle drive to propel a vehicle, said system comprising:
   - a turbine having a vaned rotor for connection to said vehicle drive and responsive to fluid under pressure to operate said vehicle drive;
   - means normally operative to continuously supply said fluid under pressure to said turbine and including a throttle valve for regulating the rate of flow of said fluid to said turbine;
   - an electric motor connected to said means for operation thereof;
   - electric batteries in circuit with said electric motor for energization thereof;
   - generating means connected to and operative by said rotor for generating electrical energy, said generating means being coupled in said circuit with said batteries;
   - a voltage regulator in said circuit;
   - and means for detachably coupling said circuit to a separate source of energy for periodically recharging said batteries.

2. A fluid pressure system according to claim **1** wherein said first-mentioned means includes an air compressor connected to said motor; and
   - a compressed air reservoir coupled to said air compressor and coupled to said turbine through said throttle valve.

3. A fluid pressure system according to claim **2** wherein said air compressor includes an air intake scoop operative to receive ram air during forward movement of said vehicle.

4. A fluid pressure system according to claim **1** and including means for conveying fluid exhausted from said turbine to said first-mentioned means.

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May 26, 1970 DAE SIK KIM 3,513,929

LOW-POLLUTING ENGINE AND DRIVE SYSTEM

Filed Aug. 25, 1967 2 Sheets-Sheet 1

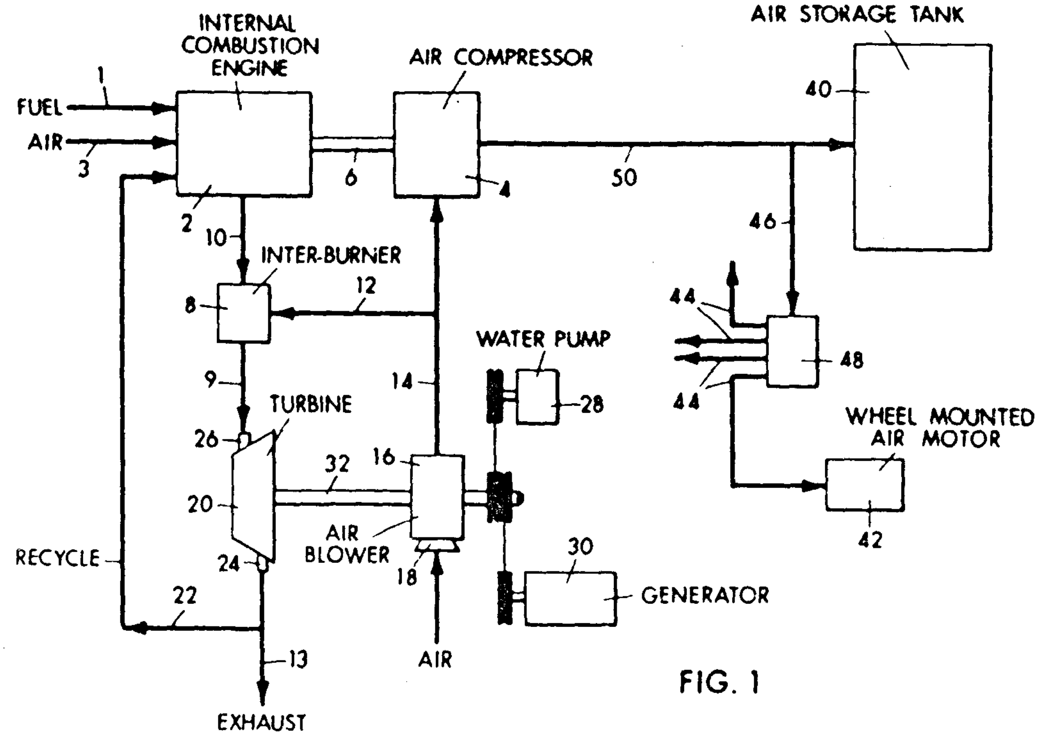

FIG. 1

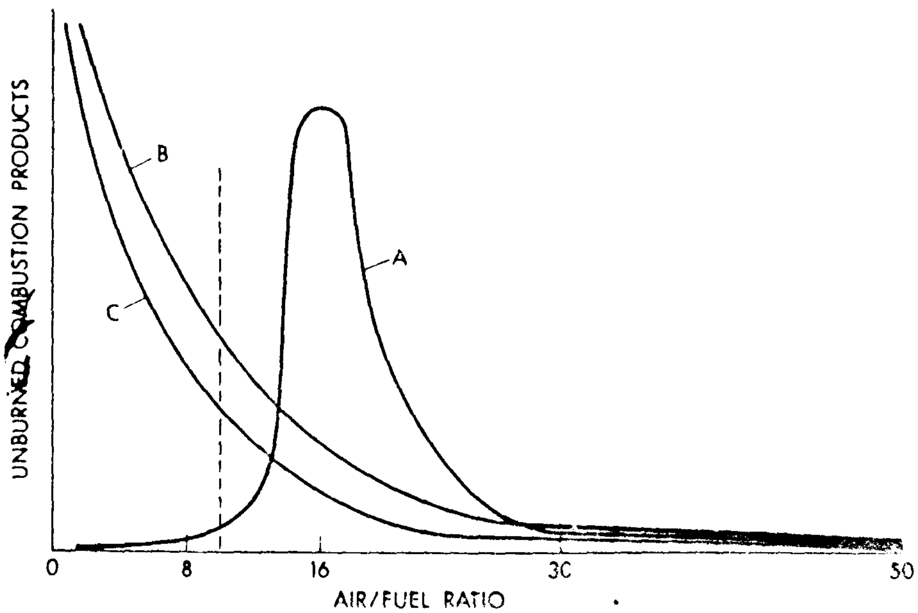

FIG. 2

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LOW-POLLUTING ENGINE AND DRIVE SYSTEM

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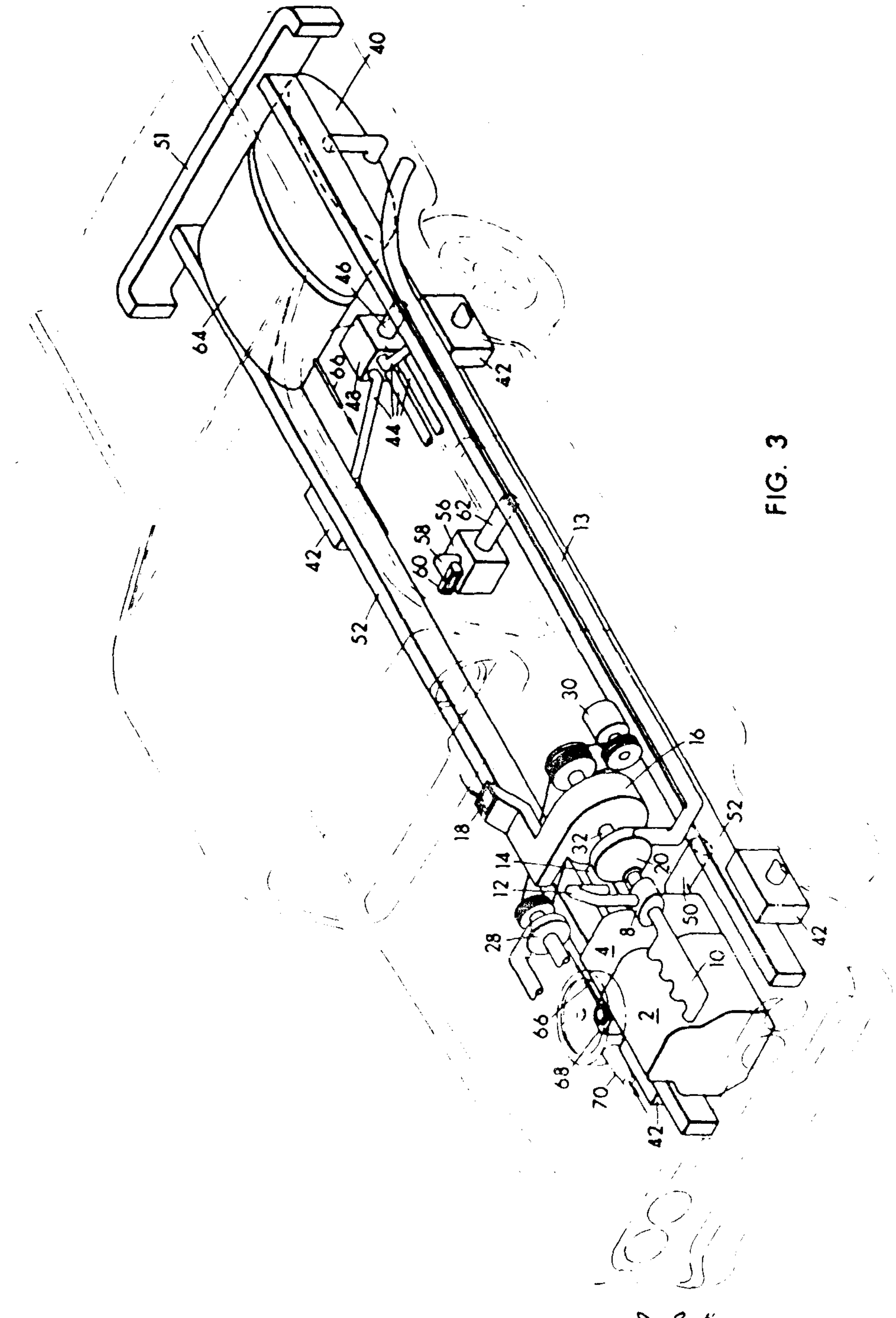

FIG. 3

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3,513,929
LOW-POLLUTING ENGINE AND DRIVE SYSTEM

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U.S. Cl. 180—66 13 Claims

ABSTRACT OF THE DISCLOSURE

A low-polluting fossil-fueled engine and drive system is achieved by utilizing in combination an internal combustion engine operatively associated with a compressor, an interburner which completes the combustion of unburned hydrocarbons and carbon monoxide from the internal combustion engine, a gas turbine which recovers power from the hot exhaust leaving the interburner and a compressed air reservoir which serves as a power storage tank and which supplies compressed air to a plurality of air motors.

FIELD OF THE INVENTION

This invention relates in general to an internal combustion engine driven power train. More particularly, it relates to an improved engine and drive system which affords high efficiencies of operation while at the same time greatly reducing the amounts of pollutants emitted to the surrounding atmosphere. In its most specific form the invention is directed to a low-polluting gasoline powered engine and drive system for use in an automobile.

During the last fifty years the high compression ratio gasoline engines used in today's automobiles have been improved almost to the point of perfection. Their most desirable performance characteristics are a quick response to level of power and their low wt./H.P. ratio. However, when exhaust emission is considered, the widely variable operating characteristics which are essential to today's directly driven automobile power systems cause difficulties in the design of an optimum exhaust elimination system. To date, various methods of the reduction of exhaust pollutants have been tried. These include, for example, the use of a catalytic afterburner and some recycling of exhaust gases. However, the methods of eliminating pollutants tried to date decrease the driveability of the vehicle; that is, losses are experienced in some of the operating characteristics of the engine and power train.

In the last five years steps have been taken to curb the air pollution due to internal combustion engines. In response to the need for reducing pollution, automobile manufacturers have started installing the so-called "blow-by" devices in most of their models for recirculating and burning gases which escaped from the cylinders passed the pistons. Many of these gases are mainly hydrocarbons that earlier models vented into the atmosphere. However, exhaust pipe gases still remain as the major source of automobile emissions to the atmosphere. These gases include carbon monoxide, nitrogen oxides and hydrocarbons.

The problems encountered in attempting to eliminate exhaust pollution may be understood from the following brief discussion. The power train of the present car is basically a direct drive system and, hence, the operation of its engine must meet all driving conditions. This in turn introduces a tremendous transient variation in the gas flow rates through the engine, in the exhaust temperature and in the exhaust composition. Any device which is attached to such an engine must also be effective under such widely varying conditions. For example, variation in air flow rate may easily be 20 to 50 fold between full throttle operation and idle with the time required for such change being on the order of one second. The difficulty of meeting such adverse requirements is manifested abundantly in the literature. It will also be appreciated that the present configuration of direct drive power trains requires inefficient part-throttle operation for the majority of the time and a large engine capacity (several hundreds of horsepower not being uncommon) which is rarely used or required.

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Newer standards for motor vehicle air pollutants are now being developed under the federal "Motor Vehicle Air Pollution Control Act of 1965." In response to this, automobile manufacturers are attempting to provide the necessary controls over such emissions by various methods, such as the one cited above and other modifications of the engine. While work is also proceeding on new drives based on fuel cells, large capacity electric storage batteries, and even solar energy systems, it will be understood that these are still in the earliest stages of development. It must, therefore, be appreciated that the burning of highly refined fossil fuels still remains and will remain for many years to come at the heart of the motor vehicle industry.

SUMMARY OF THE INVENTION

The device of the instant invention is thus directed towards achieving substantial reduction in the pollutants emitted by a fossil fuel (e.g. gasoline) powered internal combustion engine and drive system. This, according to the teachings of the instant invention, is accomplished by utilizing an interburner, a single stage turbine, an air compressor, and an air storage tank in combination with the internal combustion engine. The engine may, for example, be operated with about a 10:1 air-to-fuel ratio and the interburner then serves to complete the combustion of unburned hydrocarbons and carbon monoxide with additional air being supplied by an air blower driven by the single stage turbine. The single stage turbine recovers power from the hot exhaust leaving the interburner. The air blower further supplies compressed air to the compressor. The turbine may also be used to supply power to a generator and water pump of a conventional type.

When used in an automobile, power may be delivered to the wheels through a series of wheel-mounted air motors or a single partial admission impact turbine with a proper reduction system. These motors or the impact turbine are, of course, driven by compressed air supplied by the compressor. The compressed air tank serves as a power storage tank and supplies compressed air for peak demand, e.g. during acceleration as well as the power required for ordinary operation. When the wheel-mounted air motors are used, the engine and drive system of the instant invention eliminates the transmission, differential and drive shaft necessary in current automobiles. At the same time four wheel traction may be provided and a savings in weight and maintenance costs may be achieved. In this regard, in a preferred embodiment it is envisioned that the compressed air tank required for the system can be integrated with existing structural members in an automobile, thus achieving additional savings in weight.

It will be appreciated by those skilled in the art that the operation of the engine may be automatically controlled by the pressure of the compressed air in its storage tank. The driver would only have to control a suitable valve on the inlet of the air motors or impact turbine to regulate the air flow to and, hence, the speed of the wheel mounted air motors or impact turbine. The device of the instant invention provides better acceleration characteristics, which are limited only by the capacity of the compressed air storage tank and the size of

the air turbine while allowing the use of an optimum sized engine which may be quite small and very economical at an efficient, steady full throttle operation. Thus, aside from its low pollution aspects, the instant invention offers great advantages in that it eliminates both the need for a large engine whose peak output is rarely used and a conventional transmission. It will be understood that the system herein described is eliminating transient situations which are constantly occurring in today's high compression internal combustion engines and their drive systems during city driving and replacing them with a capacitive storage system which can furnish the transient energy demands when required.

Thus, it is an object of the instant invention to provide a low-polluting, fossil-fuel powered engine and drive system.

Another object is to provide a low-polluting gasoline powered engine and drive system for use in an automobile or other vehicle.

Yet, another object is to provide a drive system which has excellent driveability, i.e. high tractability and high peak power reserves, which can eliminate many of the standard drive components present in today's vehicles and which allows the use of smaller engines.

These and other objects as well as a fuller understanding of the invention may be had by reference to the accompanying detailed description and by referring to the drawings in which:

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a diagrammatic representation of the low-polluting engine and drive system of the instant invention.

FIG. 2 is a graph depicting the relationship between the air/fuel ratio and the amounts of various combustion products present in the exhaust of an internal combustion engine.

FIG. 3 depicts schematically the use of the instant invention in an automobile.

Referring to FIG. 1 in detail, a fuel, e.g. gasoline, and fresh air are introduced into an internal combustion engine through the lines 1 and 3 respectively. These are mixed in a carburetor (not shown) so that an air/fuel ratio in the range of 8:1 to 12:1 and preferably about 10:1 exists. The exhaust gases leaving the engine via the conduit 10 are thus rich in unburned hydrocarbons and carbon monoxide. A negligent amount of the oxides of nitrogen is also present. The gases in conduit 10 are in the temperature range from about 1800 to 2100° R. and preferably about 2000° R. and they are at a pressure in the range of from about 2 to 5 atmospheres and preferably about 4 atmospheres. From conduit 10 these hot exhaust gases enter an interburner 8 where sufficient air having a pressure in the range from about 2 to 5 atmospheres and preferably at about 4 atmospheres is introduced via the line 12 to result in an air-to-fuel ratio in interburner 8 in the range of about 30:1 to 50:1 and preferably about 40:1. The introduction of this large excess of air into interburner 8 results in drastic reductions in the amount of unburned hydrocarbons and residual carbon monoxide present in the exhaust gas without forming oxides of nitrogen. The entering air also causes a decrease in the temperature so that the gases leaving interburner 8 through the line 9 have a temperature in the range from about 1400 to about 1800° and preferably in the range of about 1400 to 1600° F. By maintaining the temperature in the range of about 1400 to 1600° F. the use of high temperature alloys is avoided in the single stage turbine 20. The gases from line 9 are conducted into an inlet 26 of the single stage turbine 20 which is operated so that the ratio of inlet pressure to outlet pressure is in the range from about 5:1 to about 2:1 and preferably in the range of about 4:1. Single stage turbine 20 recovers the energy from the hot exhaust gases and the power so recovered is transmitted via the shaft 32 to an air blower 16 and other auxiliary equipment such as water pump 28 and generator 30. Water pump 28 and generator 30 may be driven by a series of pulleys as indicated in FIG. 1. The fully expanded gases leave turbine 20 through the outlet 24 and are then exhausted to the atmosphere or partially recycled via the line 22 to internal combustion engine 2. It will be appreciated by those skilled in the art that the power level of the internal combustion engine can be controlled by properly proportioning the recycle in line 22 with the fresh ambient air. This proportioning could be easily accomplished by means of a control valve (not shown) on line 22, which is responsive to the pressure in tank 40.

As will be further discussed, the formation of nitrogen oxides in an internal combustion engine is almost unavoidable. Whenever high temperature flames are present, these oxides will be formed. However, one way of minimizing this formation is carrying out the combustion under a reducing atmosphere; that is, under fuel rich operations. This brings about predominantly those reactions which tend to consume carbon monoxide and the hydrocarbons thus starving oxygen to the nitrogen oxide reactions. Any unburned hydrocarbons and carbon monoxide formed under fuel rich conditions are eliminated as hereinbefore discussed in the interburner 8 without materially raising nitrogen oxide levels.

Returning to the discussion of FIG. 1, it is seen that the shaft 32 drives an air blower 16. Air blower 16 draws in fresh air through inlet 18 and delivers this air at an increased pressure in the range from about 2 to about 5 atmospheres and preferably at about 4 atmospheres through the line 14 to air compressor 5. It also supplies air via the lines 14 and 12 to the interburner 8 as hereinbefore mentioned. Air compressor 4 is designed to operate with inlet pressures to outlet pressures in the range 1:3 through about 1:6 and preferably about 1:5. Thus, with air entering the line 14 at about 4 atmospheres the outlet line 50 from compressor 4 will contain air in the range of about 20 atmospheres pressure. Line 50 is in communication with an air storage tank 40. Under normal operating conditions this tank is fully charged and delivers compressed air through the line 46 and the valve 48 and the lines 44 to a series of air motors (only one of which is shown) 42 or turbine which represent the final drive element. It will be appreciated that the valve 48 may be effectively used as a throttle and during periods of peak demand, for example, during acceleration, valve 48 would be opened to a substantial extent thus allowing the energy represented by the compressed air in tank 40 to be delivered to the drive turbines 42.

It may be calculated that if air tank 40 has a capacity of approximately 50 cubic feet and the air in the tank were under a pressure of about 20 atmospheres, the usable stored energy would be equivalent to about 150 H.P. minutes. To put this in its proper prospective, the H.P. minutes of energy necessary to accelerate a car weighing approximately 4000 lbs. from a standing start to 60 M.P.H. is somewhere in the range of 20 H.P. minutes. While this figure is subject to slight changes, it is more or less independent of the total time elapsed.

While a separate air compressor has been shown in FIG. 1 being driven by the internal combustion engine 2, it would be readily appreciated by those skilled in the art that these two units could be combined and in effect the net result could be achieved by the use of a free piston engine; that is, an engine having a power portion and a compression portion. In this regard, it is to be pointed out that both two and four cycle free piston engines could be advantageously and suitably used due to their efficiency in producing high pressure compressed air and due to their mechanical simplicity. It will also be appreciated that a small engine rated, for example, at 60 dynamometer H.P., which corresponds to about 40 axle H.P., could be used to achieve the same degree of driveability now ob-

tained using engines having dynamometer horsepower ratings in the range of 200–300. It has been estimated that a 4000 lb. vehicle requires 40 axle horsepower to maintain a steady speed of 80 m.p.h. Since vehicles of this type frequency have large engines often having dynamometer horsepower ratings of about 300 (producing axle horsepowers of about 200), the differential in axle horsepower is not used except during periods of maximum acceleration.

The following advantages can be obtained by employing a free piston engine in the system of the instant invention: The ability to vary the compression ratio; a low horsepower-to-weight ratio due to the elimination of the engine crank shaft and compressor drive shaft, since the compressor and engine are in a single unit; high operational flexibility and multifuel capacity (i.e. either diesel or spark ignition); low conversion loss by direct compression of air in the same cylinder and by the same piston; and independent operation of each unit. In regard to this last mentioned point, the independent operation feature of the free piston engine is particularly useful for pollutant control. For example, if a total steady lower level of 60 dynamometer H.P. is required and working with the conditions hereinbefore discussed (i.e. 10:1 air-to-fuel ratio) the turbine **20** will provide about one-third of the required power, remaining two-thirds (40 H.P.) being provided by the free pistons. Since a free piston engine can operate independently, this power can be produced by two independent 20 H.P. free piston engines rather than a single 40 H.P. unit. In city driving, only one of these 20 H.P. engines and part load operation of the turbine **20** can provide a 30 H.P. system. This power level can be reduced by higher exhaust recycle when the steady demand is low. This is a feature which the common engine cannot provide and which would allow a small city car which would be optimum from both the pollution and performance standpoints. A simultaneous two engine operation as just discussed can be automatically actuated by the pressure in tank **40**. Other advantages of this two engine system would include increased reliability (i.e. non-stalling) and its resulting safety.

Reference will now be had to FIG. 2 to explain in greater detail how the engine and drive system of the instant invention reduces exhaust pollutions. Three curves A, B and C representing respectively the amount of oxides of nitrogen, carbon monoxide and hydrocarbons present in the exhaust gases of an internal combustion engine under varying air-to-fuel ratios are shown. By referring to these curves, the manner in which the instant engine and drive system reduces the amount of pollutants will become clear. Thus, in the combustion step taking place in engine **2**, it has been indicated that the air-to-fuel ratio is in the range of about 8:1 to 12:1 and preferably about 10:1 as indicated by the dotted line. This then produces exhaust gases which have a substantial percentage of unburned hydrocarbons and carbon monoxide and a small amount of nitrogen oxides. This is in contrast to today's standard automobile engines which operate with air-to-fuel ratios somewhat higher, e.g. in the range of 12 to 15:1, which again referring to the curve would give somewhat less amounts of the carbon monoxide and hydrocarbons but would greatly increase the amount of oxides of nitrogen present. Upon exiting from the internal combustion engine, the exhaust gases of the instant invention are as hereinbefore mentioned introduced into an interburner. It will be recalled that this interburner is operated so thatt the air-to-fuel ratio is in the range of about 30:1 to 50:1. Again by referring to FIG. 2, it is seen that by operating at these high air-to-fuel ratios the amounts of carbon monoxide and hydrocarbons present are greatly reduced without materially changing the level of nitrogen oxides present. Thus, by operating the instant invention in the manner hereinbefore described, the pollutants of the exhaust gas are greatly reduced in comparison to what is currently available with today's common internal combustion engines.

Table I below presents a comparison of the amount of pollutants present in the exhaust of the instant engine and drive system as compared to those present in the exhaust of current internal combustion engines operating at their most efficient level and also under poor conditions. This table illustrates dramatically the reduction in exhaust gas pollutants which may be obtained using the device and teachings of the instant applications.

TABLE I

Pollutant	Device of instant invention	Int. combustion engine: Efficient levels	Int. combustion engine: Poor conditions
Hydrocarbons (p.p.m.)	~2	~800	1,000
Carbon monoxide, percent	Trace	4	6
Nitrogen Oxides (p.p.m.)	~20	2,000	5,000

FIG. 3 illustrates diagrammatically an automobile utilizing the engine and drive system of the instant invention. In operation fuel contained in a fuel tank **64** is fed through the line **66** to a carburetor **68** where it is mixed with air entering through the line **70**, the air-to-fuel ratio being set at about 10:1 by the carburetor **68**. This fuel rich mixture then flows into the engine **2** which either drives a compressor **4**, or in the case where a free piston engine is employed, compresses the air itself. Hot exhaust gases flow through the manifold **10** into the interburner **8** and then through the turbine **20** as hereinbefore described. The turbine **20** drives an air blower **16** through the shaft **32** and supplies the power for a generator **30** and a water pump **28** through a power train comprising a series of belts and pulleys. The exhaust from the turbine leaves through the exhaust pipe **13** which exits the gas at the rear of the car.

Blower **16** having an inlet **18** supplies air to the compressor **4** through the conduit **14** and also serves to supply air to the interburner **8** through the conduit **12**. This air in the preferred embodiment has a pressure in the range of about 4 atmospheres. Compressor **4**, which preferentially operates with an outlet-to-inlet pressure ratio of about 5:1, compresses the air to about **20** atmospheres and this compressed air is fed through conduit **50** to an air storage chamber. As illustrated in FIG. 3, this chamber may be composed in part of the structural members of the car itself. Thus, channel members **52** may be suitably formed so as to contain air at the requisite pressures. These channel members, if desired, can be in communication with bumper **51** which may also be fabricated so as to contain air. If additional air storage capacity is required, the structural members and the bumpers may be in communication with an air storage tank **40**. Thus, the instant application envisions the utilization of the various structural components present in today's automobile to provide storage space for compressed air. It is possible that the entire air storage capacity needed could be obtained by suitable use of the various structural components present in the car without the need for providing an auxiliary tank as illustrated.

During normal operation the engine **2** and, hence, compressor **4** are run at constant speed and the air tank **40** and its associated storage members, e.g. **52** and **51**, are normally filled. Thus, compressed air leaves the line **44** and is regulated by a suitable valving arrangement indicated schematically at **48** from where it is led through a plurality of lines **44** to air turbines **42** situated on the axle of each of the wheels of the vehicle. It will be understood that the vehicle can readily be designed to be a four-wheel driven vehicle or to have front wheel drive or rear wheel drive or any combination thereof. During conditons of peak demand, e.g. acceleration, the valving arrangement **48** can be automatically controlled so that full power is delivered to each of the wheel turbines **42**. Similarly, when the car is at idle and storage tank **40** is at full capacity, the power output of the engine and com-

pressor may be suitably adjusted as hereinbefore discussed.

In addition to the advantages affording a high efficiency, low-polluting engine and drive system as hereinbefore discussed, the instant system possesses the further advantage of being readily adaptable so as to provide the power for accessories which are becoming standard on many of today's vehicles. Thus, for example, the compressed air from the storage system **40** may be used to supply the energy for power brakes and power steering through the use of a suitable valving arrangement and air pistons and cylinders (not shown).

Another advantage is the simplicity with which this system may be adapted to supply air conditioning for the interior of the vehicle. Thus, if air conditioning were desired, it could readily be accomplished by simply adding a take-off line **62** from the compressed air storage and expanding this through a suitable expansion valve **56**, then leading the expanded and hence cooled air through a conduit **60** and then through an inlet manifold **58**.

Although the foregoing invention has been described in some detail by way of illustration and examples for purposes of clarity and understanding, it should be understood that certain changes and modifications may be practiced within the spirit of the invention. For example, using the same general arrangement of the internal combustion engine, interburner, and turbine (thus retaining the low-polluting characteristics of the primary power supply) the air compressor, air blower, air storage, and the air motors could be replaced by another type of energy conversion and storage means. Thus, the air compressor and the air blower could be replaced by electric generators for converting the shaft power derived from the internal combustion engine and turbine into the electrical energy. This energy could then be stored in a storage battery bank which in turn could drive a plurality of electric motors for driving the car.

What is claimed is:

1. In an automobile of the type having a free piston internal combustion engine, said engine having a power portion and a compression portion, the improvement which comprises in combination, an interburner for receiving exhaust gases from the power portion of said engine, said interburner having an outlet, a turbine having an inlet in communication with the outlet of said interburner, an air blower driven by the power recovered by said turbine for supplying air under pressure to said compressor portion and to said interburner, a compressed air reservoir for receiving compressed air from said compessor portion and a plurality of air motors driven by the compressed air from said air reservoirs for driving said automobile.

2. The combination of claim **1** further characterized in that said compressed air reservoir forms a portion of the structural members of said automobile.

3. The combination of claim **2** wherein said automobile is equipped with bumpers, said bumpers being hollow and forming part of said air reservoir.

4. A process for obtaining power using a gasoline fuel which comprises the folowing steps in combination:
   (a) injecting a fuel rich mixture having an air-to-fuel ratio in the range of 8:1 to 12:1 into the combustion chambers of an internal combustion engine, thereby producing an exhaust gas rich in unburned hydrocarbons and carbon monoxide;
   (b) introducing said exhaust gas into an interburner and completing the combustion of said hydrocarbons and carbon monoxide by introducing excess air into said interburner whereby an air-to-fuel ratio of 25:1 to 50:1 is obtained in said interburner;
   (c) passing the hot gases from said interburner through a gas turbine whereby power is recovered therefrom;
   (d) utilizing the power recovered in step (c) to drive an air blower which supplies compressed air having a pressure in the range from about 2 atmospheres to about 5 atmospheres to said interburner and to a compressor driven by said engine;
   (e) further compressing the air from step (d) in said compressor to a pressure in the range of from about 15 to about 25 atmospheres; and
   (f) using the further compressed air to supply the driving power for a plurality of air motors or impact turbine.

5. The process of claim **4** further characterized in that said fuel rich mixture has an air-to-fuel ratio of about 10:1, the interburner is fired with an air-to-fuel ratio of about 35.1, said blower supplies air having a pressure of about 4 atmospheres and said further compressing produces air at a pressure of about 20 atmospheres.

6. The process of claim **5** further characterized in that the exhaust gas leaving said internal combustion engine is at a temperature of from about 1800 to 2100° R, and a pressure of about 4 atmospheres, the gas exiting said interburner has a temperature in the range of from about 1400° to about 1600° R. and said turbine is a single stage turbine having an inlet pressure to outlet pressure ratio of about 4:1.

7. A method for reducing pollution from an automobile having a gasoline fueled internal combustion engine which comprises the following steps in combination:
   (a) burning a fuel rich gasoline/air mixture in said engine;
   (b) further combusting the exhaust gases from said engine in a large excess of air;
   (c) recovering power from the further combusted gases resulting from step (b) in a turbine;
   (d) utilizing the power recovered in step (c) to drive an air blower for supplying the excess of air used in step (b) and for supplying air to a compressor driven by said engine; and
   (e) using compressed air produced by said compressor to supply the power to drive at least one air motor which in turn drives the wheels of said automobile.

8. A drive system which comprises in combination:
   (a) an internal combustion engine having an inlet for air and fuel and an outlet for exhaust gases;
   (b) an interburner having an inlet in communication with said exhust gas outlet, said interburner completing the combustion of unburned fuel and having an outlet for exiting hot gases;
   (c) a gas turbine having an inlet in communication with the outlet of said interburner, said turbine recovering power from said hot gases;
   (d) an air blower driven by said turbine and supplying air to said interburner;
   (e) an air compressor directly coupled to the shaft of said internal combustion engine so as to be driven thereby and adapted to receive a portion of the air exiting said air blower;
   (f) motor means adapted to be driven by the compressed air produced by said compressor.

9. The system of claim **8** which includes storage means for storing the compressed air produced by said compressor and said motor means are adapted to be driven by said stored compressed air.

10. A drive system according to claim **9** wherein said internal combustion engine is operated at an air-to-fuel ratio in the range of from about 8:1 to about 12:1 and said interburner is operated at an air-to-fuel ratio in the range of from about 30:1 to about 50:1.

11. A process for obtaining power using a gasoline fuel which comprises the following steps in combination:
   (a) injecting a fuel rich mixture into the combustion chambers of an internal combustion engine, thereby producing an exhaust gas rich in unburned hydrocarbons and carbon monoxide;
   (b) introducing said exhaust gas into an interburner and completing the combustion of said hydrocarbons and carbon monoxide by introducing excess air into said interburner whereby an air-to-fuel ratio of 25:1 to 50:1 is obtained in said interburner;

(c) passing the hot gases from said interburner through a gas turbine whereby power is recovered therefrom;

(d) utilizing the power recovered in step (c) to drive an air blower which supplies compressed air having a pressure in the range from about 2 atmospheres to about 5 atmospheres to said interburner and to a compressor driven by said engine;

(e) further compressing the air from step (d) in said compressor to a pressure in the range of from about 15 to about 25 atmospheres; and

(f) using the further compressed air to supply the driving power for a plurality of air motors or impact turbine.

12. The process of claim 11 further characterized in that said fuel rich mixture has an air-to-fuel ratio of about 10:1, the interburner is fired with an air-to-fuel ratio of about 35:1, said blower supplies air having a pressure of about 4 atmospheres and said further compressing produces air at a pressure of about 20 atmospheres.

13. The process of claim 12 further characterized in that the exhaust gas leaving said internal combustion engine is at a temperature of from about 1800 to 2100° R. and a pressure of about 4 atmospheres, the gas exiting said interburner has a temperature in the range of from about 1400° to about 1600° R. and said turbine is a single stage turbine having an inlet pressure to outlet pressure ratio of about 4:1.

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United States Patent [19]

Wagner

[11] **4,124,978**

[45] **Nov. 14, 1978**

[54] COMPRESSED AIR ENGINE

[76] Inventor: **William C. Wagner**, 47 Jamaica Ave., Plainview, N.Y. 11803

[21] Appl. No.: **608,032**

[22] Filed: **Aug. 27, 1975**

Related U.S. Application Data

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[51] Int. Cl.² **F01B 29/00**

[52] U.S. Cl. .... **60/410**; 60/412; 60/418; 91/454; 91/273; 92/127; 92/162 R; 184/11 D

[58] **Field of Search** .... 60/407, 408, 409, 410, 60/412, 370, 371, 413, 415, 416, 417, 418; 180/65 A, 66 B; 92/127, 162; 91/454, 457, 273; 184/11 D

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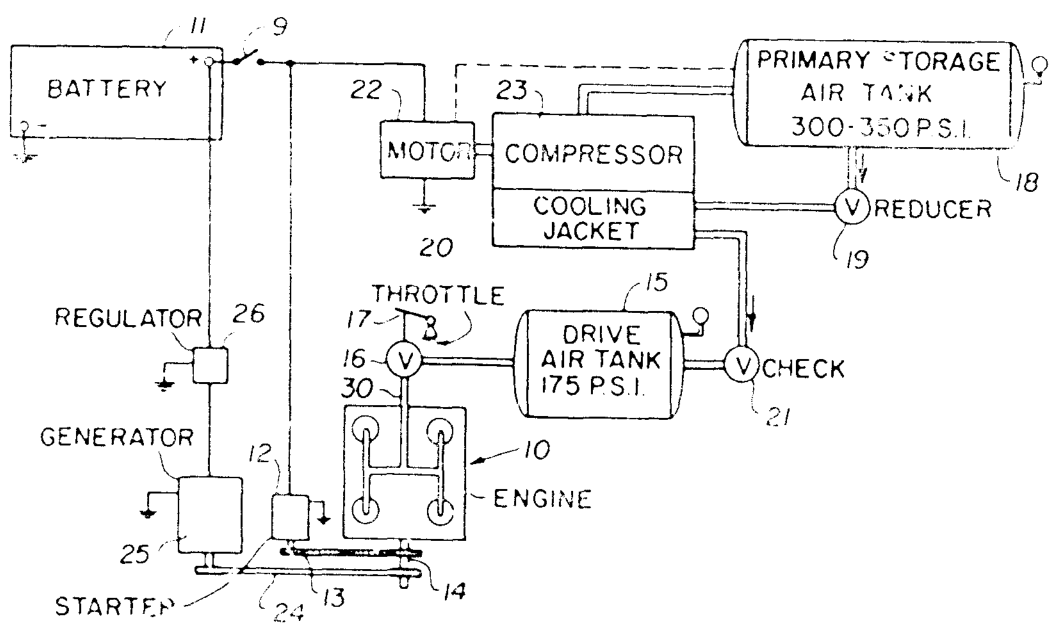
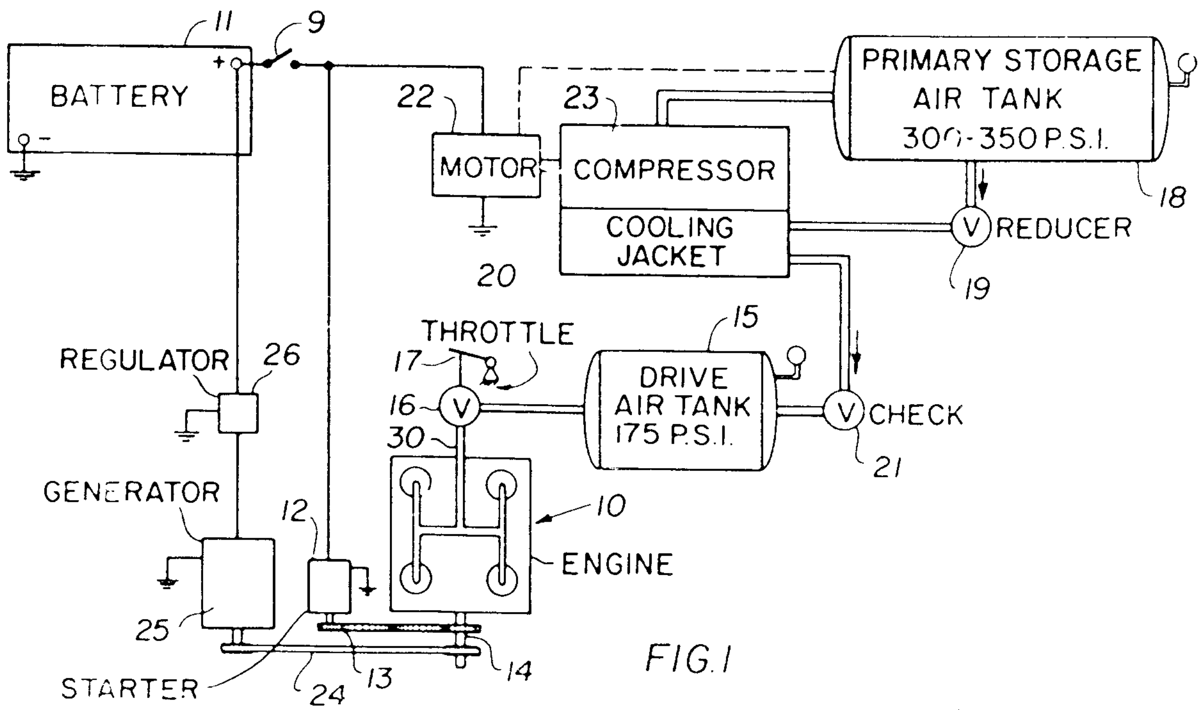
Primary Examiner—Allen M. Ostrager
Attorney, Agent, or Firm—Eisenman, Allsopp & Strack

[57] **ABSTRACT**

A compressed air engine having specially designed cylinder heads and piston configurations in order to reduce friction and maximize the application of air from a pressurized air source. The disclosure includes a lubrication system whereby the mechanical parts receive oil via a lubrication arrangement including a conveyor chain continually bathed in oil.

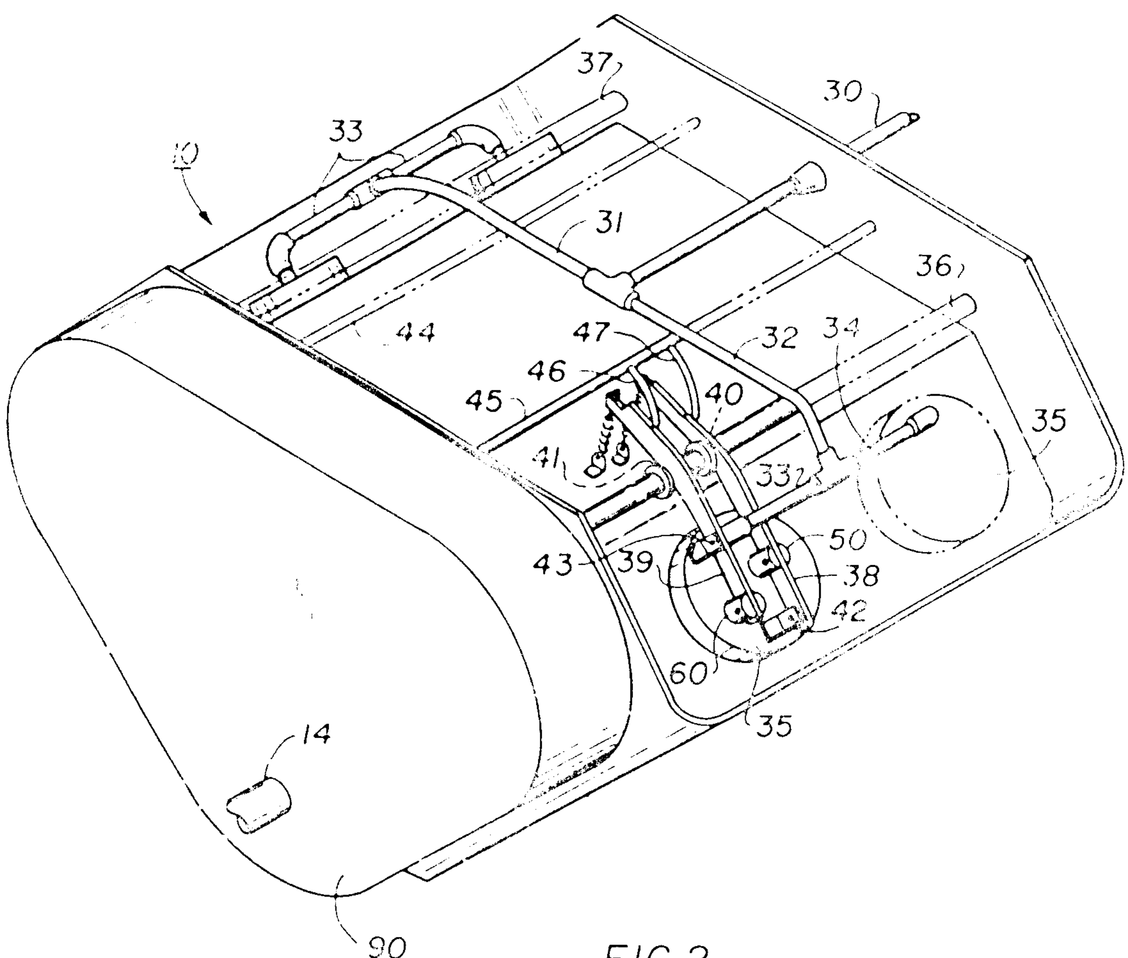
13 Claims, 7 Drawing Figures





BATTERY
11
9
PRIMARY STORAGE AIR TANK 300-350 P.S.I.
22
23
MOTOR
COMPRESSOR
COOLING JACKET
18
REDUCER
19
20
THROTTLE
15
17
REGULATOR
26
16
DRIVE AIR TANK 175 P.S.I.
CHECK
30
21
GENERATOR
10
12
ENGINE
25
STARTER
24
13
14

FIG.1



37
30
33
10
31
36
44
47
32
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46
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35
41
33
43
39
50
38
42
60
35
14
90

FIG.2

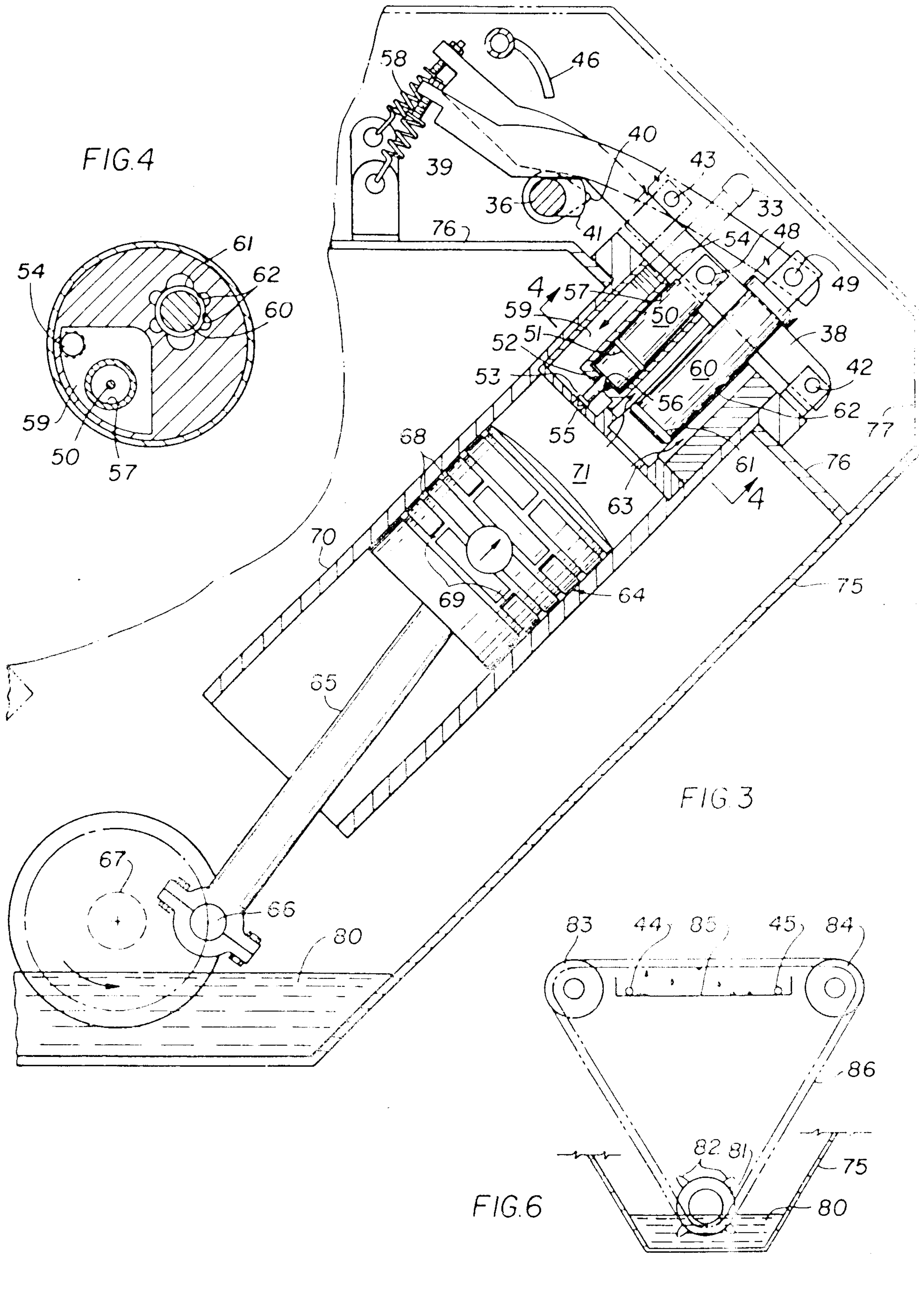
FIG.4
54
61
62
60
59
50
57
58
46
40
39
43
36
33
76
41
54
48
49
4
57
59
50
51
52
38
53
60
42
56
62
77
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68
71
76
61
63
4
70
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83
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85
45
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75
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FIG.3

FIG.6

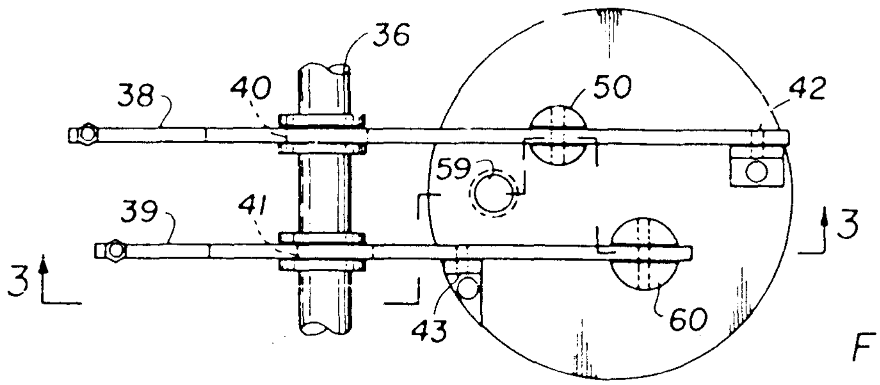
36
38
40
50
42
59
39
41
3
3
43
60

FIG. 5

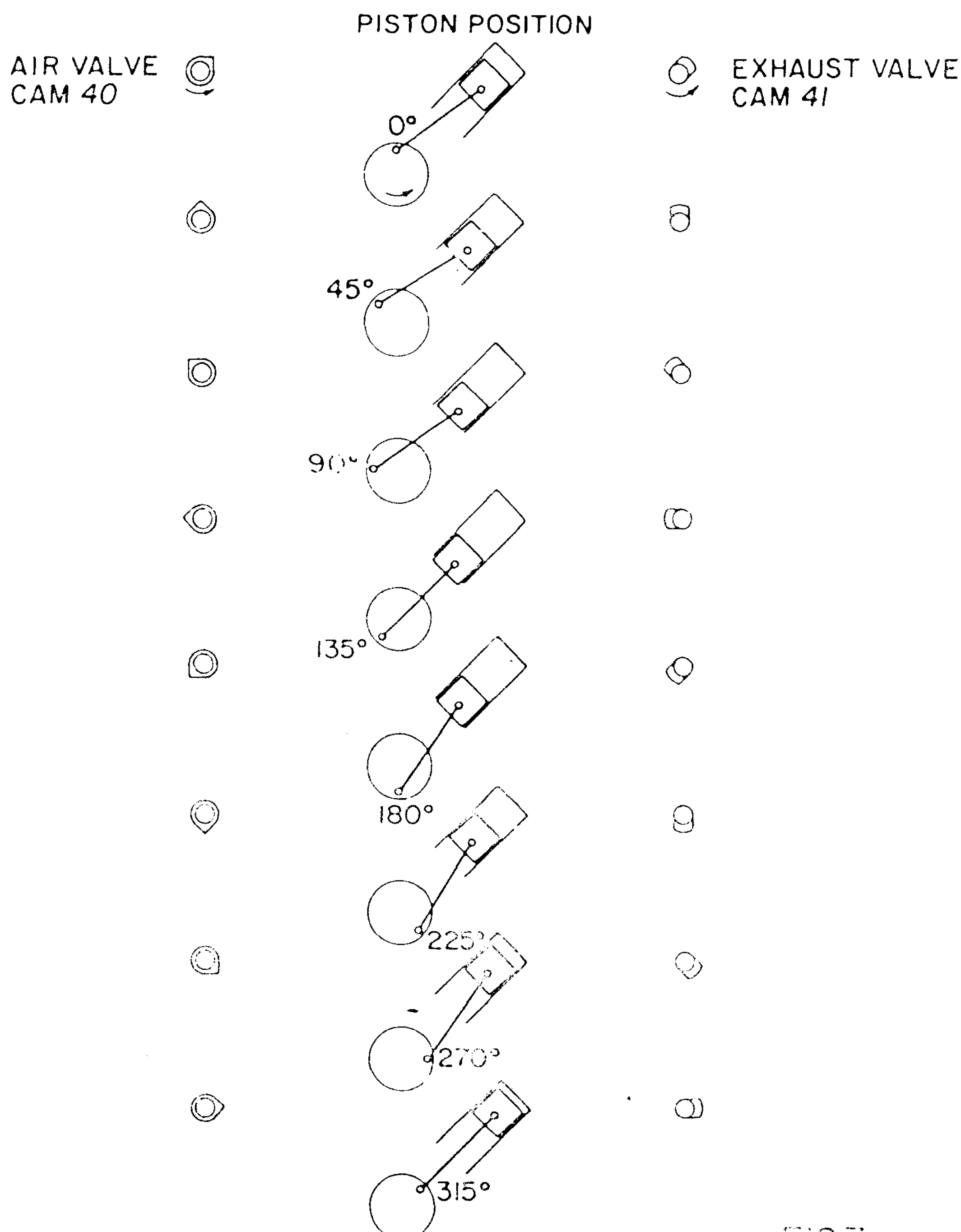
PISTON POSITION
AIR VALVE
CAM 40
EXHAUST VALVE
CAM 41
0°
45°
135°
180°
315°

FIG. 7

101050 0076 -05 11-14-78 22853

10-10-78 210554 781470 04124978 0493

COMPRESSED AIR ENGINE

RELATED APPLICATIONS

This is a continuation-in-part of co-pending application Ser. No. 473,420, filed May 28, 1974 now abandoned.

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates generally to compressed air engines and more particularly to improved compressed air engines having specially designed components which maximize utilization of the air and minimize friction.

2. Description of the Prior Art

Compressed air engines have been made in various forms for many years. Such engines have long found applicability in volatile atmospheres where the ignition of gaseous engines is intolerable. The basic components of such engines include cylinders, reciprocating pistons, means for selectively supplying air under pressure to the cylinders, and exhausting the air after extracting the pressure. It is also common practice to drive such engines from a storage source of air under pressure, which is replenished, or kept within a desired pressure range by means of a compressor.

Such engines require an external source of power in order to initiate operation in many instances, and also in order to make-up for the necessary depletion of air and drop in pressure during operation. It is essential to minimize friction in this type of engine to recycle the input energy for as long as possible, and to efficiently store any unused energy, in order to achieve the greatest performance.

Internal combustion engines have been employed in the past to produce the necessary compressed air, but the inherent atmospheric pollution of such engines is objectionable. It has been demonstrated that electrically powered compressors may be advantageously used to avoid pollution and this suggests ease of installation in fixed locations, or suitability for mobile use with rechargeable batteries.

SUMMARY OF THE INVENTION

The present invention is embodied in a multiple cylinder compressed air engine driven via a drive tank of compressed air. A primary air storage tank of greater volume than the drive tank, contains air at greater pressure than the drive tank which is maintained within a predetermined pressure range by means of a compressor driven with either a diesel engine or an electric motor. When a motor is used, an electric battery supplies current to the motor, a generator is directly driven by the compressed air engine in order to supply recharging current to the battery and alternatively provide power for the compressor, and a heavy drive shaft is employed to store energy and smooth out operation.

It is an object of the present invention to provide an improved compressed air engine.

Another object of the invention is to provide an improved compressed air engine with a high capacity for recycling any undissipated energy during normal operation.

Another object of the invention is to provide such an engine with a plurality of cylinders each having pistons mounted for reciprocating motion and spaced from the cylinder walls by a cushion of air.

It is another object of the invention to provide an improved compressed air engine having uniquely designed cylinder heads in order to maximize the input and exhaust of air to and from the cylinders.

In accordance with a particular embodiment of the invention there is provided a compressed air engine including a source of air under pressure, and at least one cylinder assembly comprising: a cylinder, a piston mounted for reciprocating motion in the cylinder, and a cylinder head having input and output valve means. The input valve means include an auxiliary chamber interposed between the source of air under pressure and the cylinder. The input valve means also includes a valve member operative to cyclically admit air from the auxiliary chamber to the cylinder in order to drive the piston.

In accordance with another aspect of the invention, there is provided a piston structure including circumferential grooves encircling the piston walls and axial grooves interconnecting the circumferential grooves. The wall surfaces of the pistons are dimensioned for slight clearance within the cylinder. Thus, upon introduction of air through the cylinder head, the major force of the air acts upon the piston face, but a small portion of this air traverses the grooves in the side walls of the piston. As a result, the piston in effect, floats free within the cylinder and in so doing reciprocates without frictional contact against the cylinder walls.

In accordance with yet another aspect of the invention, a compressed air engine embodying the features of the invention is provided with a complete lubricating system. A principal component of the lubricating system is a continuous chain driven by the drive shaft and immersed in an oil bath at the lowermost portion of its travel. The lubricating chain deposits oil at the upper portion of its travel into a drain pan connected by oil ducts or conduits to each of the major frictional points of the engine.

A more complete appreciation and understanding of the invention will be available from the following text and the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a block schematic showing the principal components of a compressed air engine embodying the invention;

FIG. 2 is a perspective illustration of a compressed air engine embodying the invention and showing typical components and their relative positions;

FIG. 3 is a vertical cross-sectional view taken along the lines 3—3 of FIG. 4, illustrating the structure of a typical cylinder head embodying the features of the invention;

FIG. 4 is a cross-section taken along lines 4—4 of FIG. 3;

FIG. 5 is a top view showing the cams, cam followers and cylnder head of a typical cylinder embodying the features of the invention;

FIG. 6 is an illustration of the oil lubrication system located behind the front cover of the engine shown in FIG. 2; and

FIG. 7 is diagrammatic chart showing the approximate position of the cams and the piston associated with a typical cylinder throughout a complete operating cycle of an engine embodying the features of the invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIG. 1 shows the interconnections and relationships of the principal components of the compressed air engine of the preferred embodiment. The engine block 10 appears in the lower central portion of the FIGURE. Switch 9 controls the engine and is closed to establish an electric circuit from battery 11 to starter motor 12. Starter motor 12 is coupled via gears 13 to the crank shaft 14 of the engine in a manner similar to that customary with conventional internal combustion engines. Operation of the starter motor 14 is effective to turn over the engine and thereafter it continues running as long as air under pressure is applied. Once the engine is started, starter motor 14 is disengaged by means, for example, of a conventional centrifugal clutch.

Drive tank 15 supplies air to the cylinders. It has been found desirable to operate this drive tank at a constant pressure of approximately 175 lbs. per square inch. The air from tank 15 is applied through throttle valve 16 to an input conduit 30. Valve 16 controls the amount of air supplied to the engine in accordance with the position of a throttle lever 17; the linkage and operation being advantageously similar to that found on the gas pedal for conventional internal combustion engines.

Drive tank 15 is maintained at pressure by a primary storage tank 18 which contains air under a pressure which ranges in a particular embodiment between 300 and 350 lbs. per square inch. Air from the primary storage tank is applied through a pressure reducer valve 19 and cooling jacket 20 to a check valve 21 at the input of the drive tank 15. Cooling jacket 20 may not be essential; however, the inherent cooling available as a result of the expansion of the air as it comes out of reducer valve 19, can be used to reduce the heat developed within compressor 23.

The pressure of the air within the primary storage tank 18 is maintained within the desired range by a compressor 23 that is driven by electric motor 22. The compressor motor 22 is directly connected to battery 11 via switch 9 and is energized whenever the pressure in the primary storage tank falls below 300 lbs. per square inch. The pressure actuated enabling switch for motor 22 is not illustrated in the drawings; however, such switches are familiar to those skilled in the art. Alternatively, motor 22 may be replaced by a diesel engine that is completely independent of the compressed air engine. This diesel engine will simply operate to maintain the pressure within prescribed ranges.

In order to reduce battery drain, a generator or alternator 25 is directly coupled to the crank shaft 14 via a pulley and belt system 24. The output of generator 25 is connected through regulator 26 in order to supply charging current to battery 11, as required. It will be understood that compressor 23 is driven by motor 22 only during those times that the pressure in the primary storage tank is below 300 lbs. per square inch. Generator 25 effectively supplies charging current to battery 11, as is in parallel with the battery in supplying the compressor motor when it is energized. The capacity of compressor 23 and generator 25, and the volume of primary storage tank 18 and drive tank 15, are selected in accordance with the load to be driven by the compressed air engine and the operating periods contemplated.

The use of crank shaft energy to assist in battery recharging and compressor operation, may be likened to the centrifugal energy storage units and used successfully in some bus and trolley systems. Clearly, if the present engine is installed to drive mobile units, under some conditions (e.g. going down hills) positive energy input will be available from the crank shaft to the engine. Obviously, frictional forces will deplete the energy initially supplied, either by electrical means, compressed air, or diesel fuel input; however, it has been found that the described system does afford advantages in this type of compressed air engine.

The principal components on the engine block 10, are visible in the perspective view of FIG. 2 which reveals the front, top, and upper right side of the block, with the front cover 90 in place, and the top cover 77 removed. For simplicity of illustration, a 4 cylinder V-type engine is depicted. The cams, cam followers, and cylinder head for a single cylinder only, are shown in detail.

The air supplied by conduit 30 is divided at a T-joint into conduits 31 and 32 for distribution to the cylinders on the opposite sides of the block. On each side of the block, the air is distributed via separate conduits, e.g. 33, 34, individual to each cylinder. A camshaft 36, 37 extends along each side of the engine. Cams, e.g. 40, 41, are mounted upon the camshafts and controls the positions of cam followers e.g. 38, 39, respectively. The cam followers 38, 39 are pivotally coupled at the cylinder head and determine the relative position of the input and output valve means 50, 60.

FIG. 2 also illustrates a segment of the oil distribution system designed to lubricate the cams and cam followers. Thus, tubing 44 and 45 will be seen extending along each side of the engine. Distribution points are available at each cylinder as illustrated on the first cylinder by drain tubes 46, 47 disposed over cam followers 38, 39. FIG. 6 shows the manner in which oil is supplied to tubing 44 and 45 and this will be discussed in detail hereinafter. It may be noted in passing that all oil distributed to the upper portion of the engine will be collected within the housing and returned to an oil bath at the bottom front of the block, from which it is recirculated.

The cross-sectional views of FIGS. 3 and 4 show the interior of the typical first cylinder located on the right hand side of the engine block. This cylinder is positioned within the lower housing 75 and the cylinder head is supported upon upper housing plate 76. An upper housing cover 77 is illustrated in phantom outline.

The principal components of the cylinder assembly include the cylinder 70, piston 64 mounted for reciprocation within the cylinder, auxiliary air input chamber 59, the input valve assembly 50-57, and the exhaust valve assembly 60-63. Piston 64 is connected by rod 65 and coupling 66 to the crank shaft 67 in conventional manner for the conversion of the reciprocating motion of the piston to rotary motion of the shaft.

Air is supplied under pressure to input port 54 of auxiliary input chamber 59 by conduit 33. Chamber 59 extends axially throughout approximately one quadrant of the cylinder head. The input valve assembly occupies a separate cylindrical passage 57 within the auxiliary input chamber. The input valve assembly includes valve element 50; O-ring seal 51 bearing against the walls of passage 57; seal 53 on the end of valve 50 for seating within valve aperture 55; and O-ring 52 embedded within guide aperture 56 to guide and seal the valve stem against escape of air. The position of the input valve is controlled by cam lever 38 which is connected

to the valve at coupling **48** and is also pivotally connected at **42** to the block. The opposite end of cam lever **38** is secured to the block via a spring **58**. Between pivot **42** and spring **58**, cam follower **38** bears against cam **40** which is mounted upon camshaft **36**. The configuration of cam **40** and follower **38** is such that valve **50** is opened selectively to admit air when piston **64** is near the upper portion of its stroke. The particular timing and operation of the valve will be described in more detail in connection with FIG. 7.

Exhaust valve **60** is disposed within a fluted cylindrical channel **62** in the cylinder head. An O-ring **61** is provided at the lower extremity of valve **60** for sealing the cylinder exhaust aperture **63**. The valve element **60** reciprocates in sliding contact with the inner surfaces of fluted cylindrical channel **62**, and the fluted portions provide for the passage of air, such that when element **60** is in the retracted position shown, air is exhausted along the paths shown by the arrows. As illustrated, piston **64** is completing its upward stroke. In a short while, this stroke will be fully completed and exhaust valve **60** will close. This will be seen to result as camshaft **36** rotates in a counterclockwise direction.

To reduce friction within the cylinder, piston **64** is provided with grooves or slots **66, 69** on its outer surface. These grooves permit the escape of small amounts of air around the entire circumference of the piston. This air acts to keep piston **64** in a floating condition so that there is either no frictional contact or minimal frictional contact, with the interior wall of cylinder **70**. In particular, piston **64** is provided with circumferential grooves **68** extending about its entire perimeter. Axial grooves **69** interconnect each of the circumferential grooves and they are staggered on successive sections in order to avoid short circuiting of the air flow past the piston.

FIG. 7 is arranged with the rotational position of the air valve cam **40** depicted in the left-hand column, the position of piston **64** vis-a-vis the rotational position of the crank shaft **67** in the central column, and the position of the exhaust valve cam **41** in the right hand column. Succeeding rows in this FIGURE suggest successive drive shaft rotational positions starting at an arbitrary 0° and proceeding in 45° increments.

The first row in FIG. 7 illustrates the 0° position where piston **64** has just passed top dead center and is beginning to move downward. Air valve **50** is beginning to open and exhaust valve **60** has closed.

In the 45° position of the drive shaft, air valve **50** is fully opened and exhaust valve **60** remains closed. Air is consequently admitted under pressure into the chamber **71** formed by the upper portion of the cylinder and the piston face. This pressure forces the piston downward and in turn applies torque to the drive shaft.

In the 90° position of the crank shaft, air valve **50** closes. Exhaust valve **60** remains closed. The air valve will continue in this closed condition until the drive shaft has completed rotation and piston **64** has again begun to descend. This is pictorially illustrated in the remaining segments of FIG. 6.

At the 135° position, piston **64** begins its upward motion, and exhaust valve **60** is opened. During the entire upward portion of its travel, the exhaust valve is kept in its opened condition. Shortly before air is admitted, i.e. at approximately 45°, exhaust valve **60** is again closed. The particular timing at which the various valves open and close is of course controlled by the configuration of cams **40** and **41**.

Yet another feature of the present invention relates to a lubrication system in accordance with which the liquid lubricant, is distributed from a reservoir at the bottom of engine housing **75** to the various cams and upper engine surfaces. The partial front elevation view of FIG. 6 shows the front of the engine with the front cover removed. As illustrated schematically in this FIGURE, a chain **86** is engaged through gear **81** to be driven by crank shaft **67** and is trained over two idler gears **83, 84**. A fan-like member **82** is coupled to the drive gear **81**, and as the chain dips into reservoir **80** the lubricant is forced into direct contact with it. As chain **86** traverses the upper portion of its travel between idler gears **83** and **84**, the lubricant drains off into distribution tray **85** from whence it is distributed via conduits **44** and **45** to selected points on the engine. Weep holes are provided in the engine housing to permit all oil to regain its original position within reservoir **80**.

Engines embodying the features of this invention are suitable for a variety of uses. While a four cylinder V-type model has been constructed and described herein, other configurations are possible and in particular situations may be desirable. These engines may be employed as prime movers for vehicles or for stationary applications. The specific use of the equipment will dictate the specifications for the battery, compressor, air storage tanks, generator, etc.

Under no circumstances, should one consider these engines to be capable of perpetual motion. However, by utilizing the structures described herein, an improved compressed air engine is available for industrial and commercial use.

A particular embodiment of the invention has been shown and described. Various specific features of the invention regarding the unique cylinder head configuration, drive arrangement, and lubrication system have been detailed. Modifications will become immediately apparent to those skilled in the art. Any such modifications within the spirit and teachings of this invention are intended to be covered by the following claims.

What is claimed is:

1. A compressed air engine having a source of air under pressure, comprising: an electrically driven compressor for maintaining said pressure above a predetermined level, an air storage tank, means for supplying air from said source to said storage tank to maintain the pressure in said tank at a desired level below said predetermined level, at least one cylinder having a reciprocating piston therein, means for selectively supplying air from said tank to said cylinder to drive said piston, a crank shaft coupled to said piston and rotatably driven responsive to the reciprocating motion of said piston, means coupled to said crank shaft to supply power to said compressor, and further means operative independently of said crank shaft to supply power to said compressor, wherein said means for supplying air to said cylinder comprises: a cylinder head, an auxiliary chamber in said cylinder head, conduit means for connecting said tank to said auxiliary chamber, and input valve means operative to periodically admit air from said auxiliary chamber into the chamber formed by said cylinder head and the top of said piston, the periodicity of said admission of air being synchronized with the rotation of said crank shaft.

2. A compressed air engine in accordance with claim **1**, wherein said auxiliary chamber is disposed within an axially extending sector of said cylinder head; said input valve means includes a reciprocating valve element in a

sealed cylinder positioned within said auxiliary chamber; an aperture permits air flow between said auxiliary chamber and the chamber formed by the cylinder head and the top of the piston; said valve element sealingly seats in said aperture; and cam means are coupled to said crank shaft to control operation of said valve element.

3. A compressed air engine in accordance with claim 2, wherein said auxiliary chamber occupies approximately one quadrant of the cross-section of said cylinder.

4. A compressed air engine in accordance with claim 1, further comprising: exhaust valve means in said cylinder head operative to periodically exhaust air from the chamber formed by said cylinder head and the top of said piston, the periodicity of said exhaustion of air being synchronized with the rotation of said crank shaft.

5. A compressed air engine in accordance with claim 4, wherein said exhaust valve means comprises a fluted axially extending channel terminating in a circular aperture of diameter substantially equal to the internal diameter of said channel, and a reciprocating valve element within said channel adapted to seal said circular aperture when seated therein.

6. A compressed air engine in accordance with claim 5, wherein said auxiliary chamber is disposed within an axially extending sector of said cylinder head; said input valve means includes a reciprocating valve element in a sealed cylinder positioned within said auxiliary chamber; an aperture permits air flow between said auxiliary chamber and the chamber formed by the cylinder head and the top of the piston; said valve element sealingly seats in said aperture; and cam means are coupled to said crank shaft to control operation of said valve element.

7. A compressed air engine in accordance with claim 1, wherein said piston has an outside diameter slightly less than the inside diameter of said cylinder, a plurality of axially spaced circumferential grooves encircle the piston walls, and axially extending grooves interconnect adjacent circumferential grooves, whereby a small amount of the air admitted into the chamber formed by the cylinder head and top of said piston exits via said grooves and maintains a separation between the piston and cylinder walls.

8. A compressed air engine in accordance with claim 7, wherein said means for supplying air to said cylinder comprises: a cylinder head, an auxiliary chamber in said cylinder head, conduit means for connecting said tank to said auxiliary chamber, and input valve means operative to periodically admit air from said auxiliary chamber into the chamber formed by said cylinder head and the top of said piston, the periodicity of said admission of air being synchronized with the rotation of said drive shaft.

9. A compressed air engine in accordance with claim 8, further comprising: exhaust valve means in said cylinder head operative to periodically exhaust air from the chamber formed by said cylinder head and the top of said piston, the periodicity of said exhaustion of air being synchronized with the rotation of said crank shaft.

10. A compressed air engine having a source of air under pressure, comprising: an electrically driven compressor for maintaining said pressure above a predetermined level, an air storage tank, means for supplying air from said source to said storage tank to maintain the pressure in said tank at a desired level below said predetermined level, at least one cylinder having a reciprocating piston therein, means for selectively supplying air from said tank to said cylinder to drive said piston, a crank shaft coupled to said piston and rotatably driven responsive to the reciprocating motion of said piston, means coupled to said crank shaft to supply power to said compressor, and further means operative independently of said crank shaft to supply power to said compressor, wherein said means for supplying air to said cylinder comprises: a cylinder head, an auxiliary chamber in said cylinder head, conduit means for connecting said tank to said auxiliary chamber, and input valve means operative to periodically admit air from said auxiliary chamber into the chamber formed by said cylinder head and the top of said piston, the periodicity of said admission of air being synchronized with the rotation of said crank shaft, further comprising a reservoir containing lubricant located at the bottom of said engine, a lubricant conveyor traversing a path through said reservoir and driven by said crank shaft, a pan disposed below an upper portion of said path and above the major components of the engine for receiving lubricant from said conveyor and conduit means connected to said pan for gravity distribution of said lubricant to selected locations.

11. A compressed air engine in accordance with claim 10, wherein said piston has an outside diameter slightly less than the inside diameter of said cylinder, a plurality of axially spaced circumferential grooves encircle the piston walls, and axially extending grooves interconnect adjacent circumferential grooves, whereby a small amount of the air admitted into the chamber formed by the cylinder head and top of said piston exits via said grooves and maintains a separation between the piston and cylinder walls.

12. A compressed air engine in accordance with claim 11, wherein said means for supplying air to said cylinder comprises: a cylinder head, an auxiliary chamber in said cylinder head, conduit means for connecting said tank to said auxiliary chamber, and input valve means operative to periodically admit air from said auxiliary chamber into the chamber formed by said cylinder head and the top of said piston, the periodicity of said admission of air being synchronized with the rotation of said drive shaft.

13. A compressed air engine in accordance with claim 12, further comprising: exhaust valve means in said cylinder head operative to periodically exhaust air from the chamber formed by said cylinder head and the top of said piston, the periodicity of said exhaustion of air being synchronized with the rotation of said drive shaft.

* * * * *

“There is nothing abnormal to an efficiency greater than 1, when reheating is used; this will occur (regardless of pipe and other friction) whenever the temperature of reheating is higher than the temperature of compression.”

(*Modern Machinery,* January 1899, "The Two-Pipe System of Air Compression", A. E. Chodzko, p. 11)

AMERICAN MACHINIST

April 28, 1898.

Practical Application of High Range Compressed Air Power Transmission.

BY FRANK RICHARDS.

I use the term high range to designate the system here referred to for convenience, as it seems to fit the case, whether it has been used before or not. The system is called also the two-pipe system, on account of the return pipe made necessary to convey the air which has done its work back to the compressor, the exhaust pressure being constantly maintained very much higher than that of the atmosphere. The system has been put into practical operation in California and elsewhere in the far western portion of the United States, being known there as the Cummings sytem of compressed air transmission, the apparatus there employed being under United States patents granted to Charles Cummings, of Oakland, Cal.

Under this system the compressor in full operation receives the air returning from rock-drills, pumps, hoists, motors, or whatever it is employed to drive, at a pressure considerably above that of the atmosphere, compresses it to a still higher pressure, and then sends it out again to do more work. The operation as thus stated is as simple as that of the ordinary air compressor, which takes in free air, or air at atmospheric pressure, and sends it out again at a pressure of, say, 6 or 7 atmospheres. In compressing from 1 to 7 atmospheres or in compressing from 7 to 14 atmospheres there is no essential difference in the operation involved.

If the compressor is to work at anything like the pressures last indicated there must, however, be some provision for first charging the system to the high initial pressure required, and also some means of governing or controlling the speed and output of the compressor according to the rate at which the air may

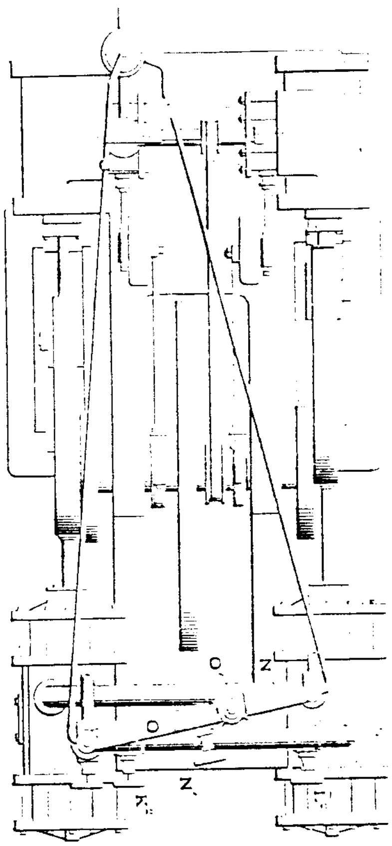

FIG. 3.—PLAN OF HIGH RANGE AIR COMPRESSOR.

be used at the other end of the system. The arrangements by which these results are accomplished are worth looking into. Fig. 1 is a side elevation, Fig. 2 is an end elevation and Fig. 3 a plan of a duplex steam driven compressor. Of the general design of the compressor it is not necessary to speak. It will scarcely be maintained that it represents the best possible arrangement. The steam cylinders are at the left, Fig. 1, and the pull and thrust of the steam piston is transmitted directly through the yoke in which the connecting rod plays to the air cylinder. The body of the air cylinder is water jacketed and the valve chambers on each end have each a vertical partition, the inlet valves and pipes being on one side and the discharge valves and pipes on the other. The air being at normal atmospheric pressure throughout the apparatus we may start the compressor, referring principally to Fig. 2, and as we proceed the functions of the pipes and connections will reveal themselves. Both compressing cylinders are alike in construction and operation. At the beginning of operations we may assume the globe valve *M* to be closed and all other valves to be open. The free air will enter through valves K^2 and L^2 and the compression will continue until the pressure rises to, say, 100 pounds, if we assume that pressure for the lower limit of our working range. When the pressure of 100 pounds is reached valve N^1 is closed by hand, separating the high pressure and the low pressure pipes, and the operation of compression then continues until the high pressure pipes are filled to the high pressure required, say 200 pounds, while the low pressure system remains charged at 100 pounds. The air inlet valves K^2 and L^2 are closed by hand and valve *M* is opened, allowing the 100 pound air to be delivered to both cylinders. The entire apparatus is now a closed system, with both the high and the low pressures as required, and ready

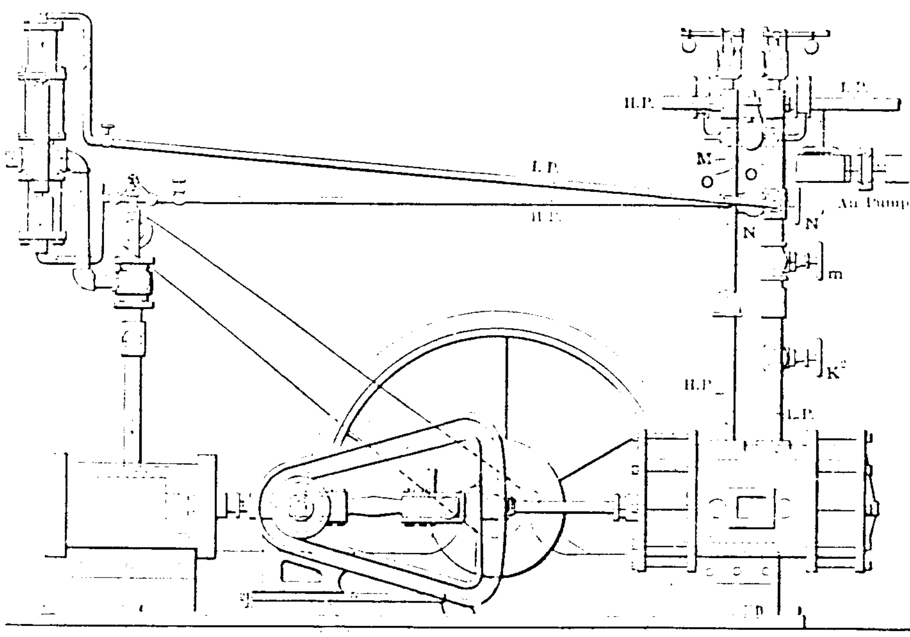

Fig. 1

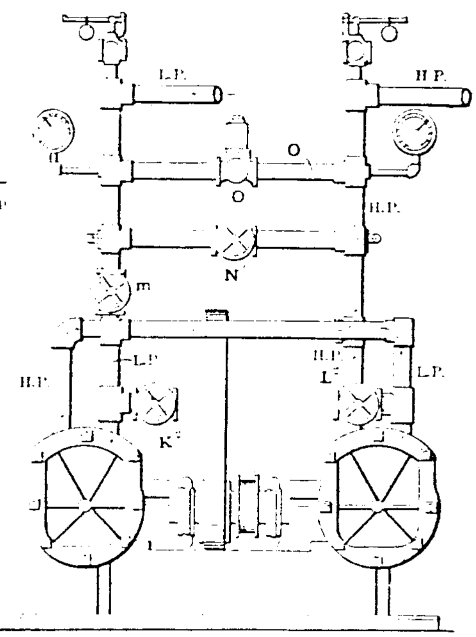

Fig. 2

SIDE AND END ELEVATION OF HIGH RANGE AIR COMPRESSOR.

for continuous action. A small pump C, shown in Fig. 1 up at the right, replaces what air may be lost by leakage. This pump may be placed wherever most convenient and may be operated by connection with the main compressor or otherwise, and here requires no further mention.

Attention is now called to cross pipe O. This pipe directly connects the high pressure and the low pressure pipes. Somewhere between the two pipes is located the valve O'. This is an adjustable pressure valve which may be set to open at any predetermined difference of pressure, in this case 100 pounds, and it will not open until that difference exists, when any excess of air in the high pressure pipes is automatically discharged into the low pressure, thus constantly maintaining the required difference. The compressor may be charged or put in condition for continuous operation by the use of this automatic valve instead of by the hand operated valve N'.

The operation of governing an air compressor, either under the system we are considering or any other, is distinctly different from that of governing a stationary steam engine. The amount of power required may fluctuate in either case, but in that of the steam engine it is still usually necessary to maintain a constant rate of speed. In the case of the air compressor it is necessary to vary the speed according to the work. With our closed circuit of piping, and assuming the compressor at one end and the motor at the other, each running at such relative speeds that the difference of 100 pounds is just maintained between the two pipes, if it should then happen that the motor should be run slower, or in any way should require less air to pass through it, then if the compressor still maintained its speed the pressure in the high pressure pipes would become too great, while in the low pressure pipes the pressure would fall too low. The reverse operation would result as badly in the opposite direction. A centrifugal governor is provided, as shown, to control the speed of the compressor when the demand upon it is up to its full capacity and prevent it from running away. This governor may also come into play in the earlier stages of the initial charging of the system, but is of no use at other times, the differential pressure governor being the usually operative controlling device. This governor, which it is scarcely necessary to describe in detail, is indicated in outline up at the left of Fig. 1. It consists of two cylinders of different areas, say one twice that of the other, the two pistons being connected and moving together and by their movement operating a sliding steam valve and controlling the flow of steam to the steam cylinder. These governor cylinders standing vertical, and that of the larger area being above, with the smaller cylinder below, a small pipe connecting with the low pressure system is led into the top of the large cylinder, while a pipe from the high pressure system enters the bottom of the lower and smaller cylinder. The differences in pressures being compensated by the differences of piston areas, the piston and the steam valve are stationary when the difference in pressures is normal, while if the high pressure increases the steam valve closes and slows the compressor, and if the high pressure falls while the low pressure increases then the steam valve moves in the other direction, more steam is admitted and the compressor runs faster. The compressor being duplex, with cranks at right angles, it will of course start itself from any position. The compressor for this system, it will thus be seen, requires little complication above that of the ordinary compressor.

It is proper to say a little as to the actual practical workings of the system. As is well known, the use of compressed air is perhaps better developed and better appreciated upon the Pacific Coast than anywhere else in the United States. Some admirable installations of the ordinary type are in operation there and some notable instances of large plants and long distance transmission. It is not strange that the two-pipe system also should find its first employment in the same section. It can scarcely be claimed for it that its possibilities have yet been all developed or that its applications have been other than crude and cheap. The half-tone herewith speaks for itself as to the conditions under which it has been tried (or it would have so spoken if the engraver had not taken such pains to make the scene respectable). The compressor here shown is of different construction from that which I have described, but it embodies the same two-pipe principle. As to the results attained, there are not as yet any thorough, careful and precise tests to be referred to, and we must be content with statements such as the following:

"Two tests were made of three hours each, one with the machinery run under the two-pipe system, and the other with the same machinery run under the ordinary system. In each one the same machinery was used throughout. This consisted of a duplex air compressor with air cylinder 5x9 inches, and steam cylinder 6½x9 inches, and a water pump with air cylinder 4½x7 inches, and water cylinder 5x7 inches." These dimensions are not altogether clear and satisfactory, but I give them as I have received them. The statement of comparative performance which follows is more intelligible.

"The machinery was run for three hours under the two-pipe, high range system. One of the pipes was then disconnected and the same machinery was run under the ordinary system. That is to say, instead of exhausting into the return pipe, the exhausting was done into the atmosphere. The pump was employed in elevating water 80 feet. The first three hours under the two-pipe system the pump made 23,877 revolutions while the compressor made 16,735 revolutions. In the second three hours, running under the ordinary system, the

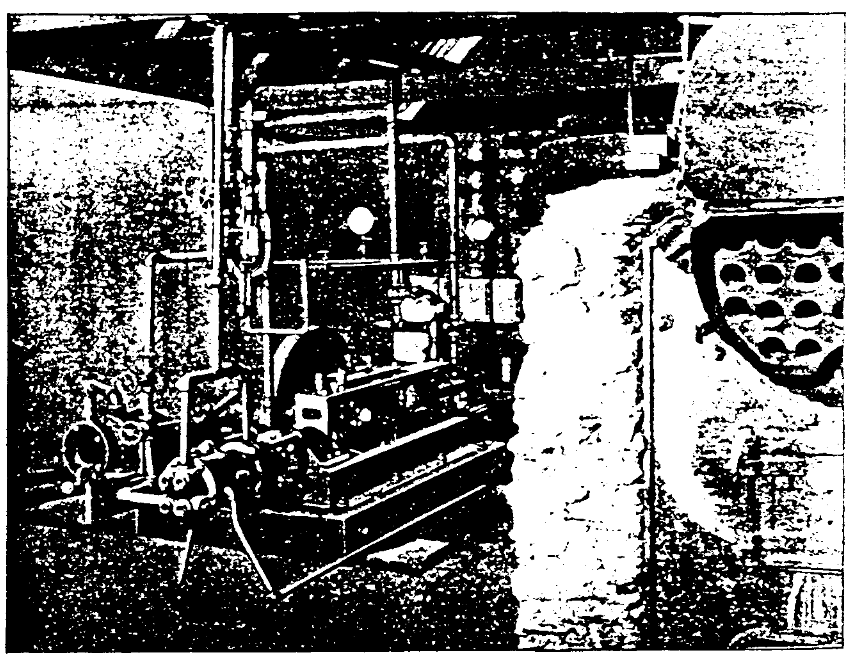

FIG. 4.—HIGH RANGE AIR COMPRESSER IN SERVICE.

compressor made 22.735 revolutions and the pump 5.380." There was, of course, no such advantage as the figures given seem to indicate, but that there was a great advantage seems clear. "In the two-pipe system we ran the pump about as fast as is consistent with good practice, while the compressor had to be run comparatively slow. In the ordinary system we could hardly run the compressor fast enough to give sufficient air to the pump to turn over. We had great difficulty in running three hours under the ordinary system, owing to the fact that the temperature in the exhaust ports of the pump came below the freezing point of water. We had to use petroleum to make the pump work. Of course, under the two-pipe system there is no chance of any ice being formed from the expansion of the air."

Hans C. Behr, in a bulletin of the California Mining Bureau, says: "The incidental advantage in the two-pipe system is that the pump engines, particularly if direct acting, can be operated under water until they wear out." He says also: "To properly estimate the value of this system it must be compared to the reheating system as to mechanical efficiency, first cost and simplicity of construction and manipulation." I have seen also enough other letters to probably convince any ordinary person that this system gives quite astonishing results.

▲ ▲ ▲

MODERN MACHINERY. January 1899

The Two-Pipe System of Air Compression.

A CONSIDERATION OF THE ADVANTAGES CLAIMED FOR IT OVER THE SINGLE PIPE SYSTEM.

By A. E. CHODZKO.

I NOTICE in the June and October issues of your paper two articles by Mr. Frank Richards, reproduced from the "American Machinist," on the so called two-pipe system of air compression and of power transmission. I deem it a good fortune to meet with an opportunity of making on this subject a more thorough inquiry than has, so far as I know, been published, upon the initiative of such an able writer on compressed air matters as is Mr. Richards.

That this system deserves more than a passing notice clearly appears from his articles; yet, in California, where it has been advocated, to the writer's knowledge, for the past ten years, being more generally known as the "Cummings process," it does not seem to have attracted much attention, notwithstanding the claim, made in its favor, of a large saving of power. The principle of this system, moreover, is not a novel one; the so called "Allen dense-air machines," which are widely used on board ship for the production of cold, have furnished for many years a successful illustration of the use of air in a closed circuit.

A complete compressed air proposition necessarily involves two distinct factors: the air is first raised to a certain pressure, and then, in this new condition, it is utilized for some specific purpose. We must therefore consider not only the compressor end of such a proposition, but the motor end as well, the economical value of the system being determined by comparing the work done in the motor with the work absorbed by the compressor. Very aptly does Mr. Richards remark that the cheaper compression of the air is as distinct a gain as would be a more efficient employment of it after compression; but when the whole system is concerned, this statement is strictly true only in so far as the utilization of the compressed air is, if not as efficient as his compression, at least not correspondingly defective.

Now, if the efficiency of the two-pipe compression is, for a given volume of air, higher than in the single pipe process, and might be theoretically increased up to 100 per cent., without interference of extraneous causes, such as reheating, the writer must confess his inability to secure some conclusive data regarding the performance of air motors in the double pipe system. He has heard of rock drilling plants of this description being used in some mines in California; and Mr. Richards quotes a few data in connection with a comparative test between the one and two-pipe systems on a pump operated by compressed air.

These results are not sufficient to give a clear idea of the general value of the latter process, a fact which Mr. Richards acknowledges, when he says: "We must be content with statements such as the following," etc.

If the two-pipe system actually resolves itself into an economical manner of producing and using compressed air, the sooner we know it the better, but if its application is attended by practical defects, it is just as desirable that they should be well understood. The following remarks are intended to throw, if possible, more light on a matter of importance to many of your readers, the writer included. With this object in view he offers what seems to him a rational view of the subject, inviting, at the same time, a criticism of his conclusions, in that spirit quite appropriately described in your paper as "Our Idea Exchange." Let those who happen to possess some practical experience of a subject under discussion give their fellow workers the benefit of what they know. This is true progressive policy, which does not interfere with the protection of individual interests, and a more liberal exercise of which would increase the stock of general knowledge, to the advantage of its adherents.

It hardly needs be stated that any intention of personal disparagement is totally absent from an article of this nature; but should it help in advertising a good thing, none would enjoy it more sincerely than the writer.

The following results are theoretical, inasmuch as taking no account of frictional losses, which will be subsequently referred to; they will serve the purpose of comparison, which is the sole object in view at present.

Three distinct cases will be considered, namely:

A. Ordinary process of compression of free air to 100 pounds effective per square inch.

B. Two-pipe process of compression from 100 pounds to 200 pounds effective per square inch.

C. Two-pipe process of compression from 1,000 pounds to 1,100 pounds effective per square inch.

The working pressure is, therefore, in all cases, 100 pounds per square inch, and the outside temperature is supposed to be 60° F.

It is also assumed that the compression is effected in one stage and within a water-jacketed cylinder, so that in the calculations referring to the compression of air, the ratio of its specific heats at constant pressure and at constant volume may be taken as 1.3.

The expansion in the motor is supposed to take place adiabatically, the above ratio being, in this case, 1.4. The following calculations refer to one cubic foot of air at initial pressure.

Under these assumptions three classes of motors will be dealt with.

I.—Motors working at full pressure during the whole stroke, such as rock drills and non-compound direct-acting pumps, the compressed air, after leaving the motor, being expanded to the initial pressure (i. e. to the pressure existing at the beginning of the compression) without doing useful work.

	A	B	C
Work of compression and of delivery into the receiver, foot lbs.	5536.1	11085.25	13919.1
Temperature (F.) of compression	375°	141°	71°
Volume of compressed air at 60° F., cubic ft.	0.128	0.534	0.911
Work of compression and delivery per cubic foot of compressed air at 60° F., foot lbs.	43250	20757	15279
Comparative efficiency of the three kinds of compression	1	2.081	2.829
Work at full pressure in motor, foot lbs.	1843.2	7689.6	13118.4
Efficiency of the system, i. e. ratio of work done in motor to work absorbed in compression	0.333	0.693	0.942
Comparative efficiency of the three systems	1	2.081	2.829

The relative efficiency is the same for the whole system, including compressor and motor, and for the compressor alone, when its work is measured per cubic foot of compressed air. This is due to the fact that the working pressure being the same in all cases, the work done in the motor also depends upon the volume of compressed air.

2. Motors working at complete expansion (i. e. the pressure resuming the value which it had at the beginning of the compression) without preliminary reheating (i. e. the compressed air being at 60° F.)

The following results apply to this case:

	A	B	C
Work of compression and delivery (as above) ft. lbs.	5536.1	11085.25	13919.1
Temperature (F.) at the end of Expansion	—172°.4	—26.4°	46.3°
Total work of Expansion (i. e., work at full pressure, followed by complete expansive work) ft. lbs.	3293.2	9493.8	13460.5
Efficiency of the system	0.595	0.856	0.967

When the working pressure is given, the higher the return pressure (meaning the lower pressure in the two-pipe system) and the smaller the difference between the work of compression and the work done in the motor. Should the return pressure become infinitely great, the efficiency of the system would become equal to 1.

3. Motors working at complete expansion with preliminary reheating.

The most economical degree of reheating should be such that the air leaves the motor at the same temperature at which it enters the compressor, the heat supplied being, in this case, completely utilized, and the successive phases occurring in the closed circuit being indefinitely repeated.

Under the present conditions, i. e. the temperature at the end of the expansion being 60° F., the following results would obtain:

	A	B	C
Temperature (F.) of reheating	447°	162°	74°
Total work of Expansion done in the motor after reheating, ft. lbs.	5927.8	11354.6	13823.9
Efficiency of the system	1.071	1.024	0.993

There is nothing abnormal to an efficiency greater than 1, when reheating is used; this will occur (regardless of pipe and other friction) whenever the temperature of reheating is higher than the temperature of compression.

The last figures show that, in what might be termed the hot expansion of air in a motor, the relative efficiencies vary in opposite direction to what they do in the cold expansion.

Here the objection might be raised that a temperature of 477° is not used in practice; this will therefore be reduced to 400°, a temperature found in high pressure steam engines and exceeded in gas and other heat engines.

Since the temperature of reheating in the two-pipe system is much lower, this reduction will first be made in the case (A) of air expanding to atmospheric pressure, with the following results:

	A	B	C
Temperature (F.) of reheating	400°	162°	74°
Efficiency of the system	0.982	1.124	0.993
Rise of temperature (F.) in eachcase	340°	102°	14°
Weight of compressed air reheated, lbs.	0.0764	0.596	5.27
Total thermal units required for reheating	6.19	14.48	17.56
Comparative amounts of heat consumed	1	2.34	2.83

And comparing the results for the same consumption of heat:

	A	B	C
Thermal units used for reheating	6.19	6.19	6.19
Temperature (F.) of reheating	400°	103°.6	64°.9
Total expansive work after reheating, foot lbs.	5440.4	10281.8	13581.6
Efficiency of the system	0.982	0.927	0.976

These three classes of motors cover the entire ground, if the air does its work in one cylinder, or more exactly, in one stage, and the above results can be summed up as follows: Whenever the air is used cold, the two-pipe system is far superior to the ordinary process of transmission; but this superiority disappears if only a moderate amount of heat is available, in which case there is but little difference between the two systems, the advantage being, however, in favor of the single-pipe method.

These conclusions are drawn from theoretical figures, and it is now expedient to examine whether some practical defects are liable to develop in either case. Considering, for instance, a direct-acting pump, which has to raise water to a certain height, using air either at

full pressure or expansively, we find that this work requires a certain number of cubic feet of compressed air per minute, at a working pressure of, say, 100 pounds per square inch, and from the above figures, it appears that a smaller compressor and a less amount of motive power will be required to supply that air in the two-pipe system than with a free air machine. The piston packings need not be tighter in either case, since the difference of pressure is the same between both sides of the piston. The stuffing-box packings will generate more friction in the two-pipe machine, but then its size is smaller, so that no apparent disadvantage seems to exist against it on that ground.

Next comes what seems to have been the most serious objection to its adoption, namely, the necessity of a double line of pipes, which, first of all, means twice more joints, half of which are under more than working pressure. Besides, and referring to the three cases previously examined, the spouting velocities of air at the higher pressure escaping into air at the lower pressure are respectively:

	A	B	C
Spouting velocity per second ft........	1563	1016	404

So that, if a certain volume at a higher pressure escapes into a capacity at the lower one, the mean velocities of flow are respectively:

	A	B	C
Feet	782	508	202
Or vary as	3.87	2.51	1

In other words, the air in the two-pipe system is "sluggish," and when acting on a piston against a back pressure, it has, aside from the useful work performed, a tendency to flow at a slower rate of speed than in the free air process. And while the above spouting velocities, even reduced by coefficients of correction, are still higher than the usual speeds adopted in machine construction, it must be remembered that the "dense" air will, at the same velocity of flow, develop more friction than air at working pressure. It seems, therefore, reasonable to surmise that the machinery operated in the two-pipe system should be especially designed in order to secure the highest efficiency; not only should the castings be made heavier to withstand the increased pressure, but the ports and passages should be shorter, straighter, and of larger area. For the same reason, the high pressure or live air pipe must be of larger size, and of course, of greater thickness than the single pipe; the same is true of the return pipe, which, even if carrying no more than working pressure, has a larger volume of air to deliver. The extra cost due to the excess of size and of weight of these pipes if the compressor is far away from the motor, may offset, if not exceed, the reduction incurred in the cost of the compressor.

Again, taking, for instance, a mine where the compressor is located near the collar of the shaft, and operates some machinery at the lower levels: the live air pipe and the return pipe are run down the shaft side by side, filled with air weighing a certain number of pounds and of ounces, and whose mass is subjected to a series of consecutive periods of rest and of motion. If the motor works at full pressure on the whole stroke, the two pipes are in constant connection through a moving piston, so that the dead weight of the air in the return pipe is balanced by a portion of the weight of live air contained in the feed pipe. But if the motor works expansively, this balance is destroyed at the moment of cut-off, and during the remainder of the stroke the motion of the air in the return pipe is produced at the expense of some power, either in the motor or in the compressor. A column of air at 100 pounds exerts a back pressure of 0.41 pounds per square inch for every 100 feet of vertical height, and these come in deduction of the mean effective pressure during the period of expansion. In a tunnel, where the pipes are on a level, or practically so, the inertia of the mass of air only is to be overcome. The amount of this loss could not be determined without taking a specific example, and a mathematical treatment of the question would be out of place in this article; but a logical idea of the facts may be gained by considering a limit case when a very high return pressure is used (which, as previously stated, increases the efficiency), and when the ratio of the air pressure in the two lines of pipe becomes very little different from the unit.

The compressor then becomes a mere displacer, imparting to the motor an action synchronous to its own, by means of a fluid transmission, a device which has been successfully applied with water as the transmitting medium; but then, a rapid motion of the operated engine is out of the question.

This, as stated, is a limit case, in which the actual conditions of work in the two-pipe air system are greatly exaggerated; but in the writer's belief, it correctly illustrates the general functions of a closed circuit, which that system implies. To what extent these elements of resistance would make themselves felt in practice depends essentially upon local circumstances, such as pressure used, length of pipes, etc.

The tightness of the joints in the two pipe lines is an important element of success, because any leak of consequence would require the refilling of the circuit by an auxiliary compressor; and while no great difficulty exists in maintaining a tight line of piping in the open air, when it is not subject to injury, the conditions are less favorable in a mine.

While the writer disclaims a personal experience of the two-pipe system, which alone would permit to express a categoric opinion, he will, in concluding, formulate as follows his impression on the subject:

If the maintenance of the double pipe line proves to be satisfactorily practicable, it is out of the question that in the majority of cases, and, namely, for underground work, a cold air machine is more desirable than a hot air motor; and the above results show that an air engine working at full pressure, or nearly so, with a liberal amount of lead, should constitute a very efficient motor, with no danger of freezing, for all kinds of purposes. The regulation and the details of construction of such a machine need not be considered at present. Both compressor and motor would be, for a given work performed, of smaller size than in the ordinary system. On the other hand, it has been shown that a hot air single pipe motor is just as efficient; but the economy in the compressing power disappears. The double-pipe system seems to be particularly well adapted to the motion of a direct acting non-compound pump, working in a damp place, where no heater could be properly installed and maintained. Concerning the rock drilling machinery, the writer believes that large sized drills and channelers, working in the open air, at a moderate distance from the compressor, might be profitably operated by this system. However, some positive evidence as to the com-

parative performance of single and double pipe rock drills would be necessary to modify his belief that, owing to the rapid changes of direction in the flow of air which occur in these machines, a single-pipe drill will deliver smarter, if not quicker, blows than a double-pipe machine, and do more work in a given time. This does not mean that this work will be done with better or equal economy.

For underground work, the single-pipe machine seems preferable, first, for the reason just mentioned; then because this machine ventilates the front, which is not the case with the two-pipe drill; and then because the frequent disconnection of joints in the high pressure pipe, easily liable to injury, from blasting or otherwise, in a narrow space, makes an extra pipe of this description undesirable.

Shortly after the foregoing was completed, the writer had the pleasure of meeting Mr. Charles Cummings, who has been for a number of years the apostle of the two-pipe system on the Pacific coast, and who kindly put at his disposal several testimonials concerning the operation of rock drills on this system.

H. P. Gray, of San Francisco, who erected and put in running order the Cummings drilling plant of the Lightner Mining Company, at Angels Camp, Col., states that he can conscientiously recommend the Cummings system for air-actuated rock drilling machinery. The consumption of wood for twenty-four hours was but two cords of bull pine. During this time the two 3¼ inch drills ran between ten and twelve hours, 9x9 hoist about ten hours, and a double acting sinker, 5x24, continuously except when shooting. The shaft is a three compartment vertical, 100 feet deep, and is in fairly good drilling ground. Regarding the drills, Mr. Gray states that the blow is hard and the return throws the sludge clear over the bar, which is about five feet from the ground. This throwing of sludge saves much time which would otherwise be spent in gunning holes. In the amount of power consumed, Mr. Gray considers the Cummings system superior to all others.

The following is taken from a letter written by Mr. Lewis F. Barlow, a San Francisco mining contractor: "I am running one of your 3¼ inch Cummings drills with a Cummings compressor, double pipe system, in a tunnel at the Summit Mine, Sunny Hill, Cal., and desire to express my complete satisfaction with the same. The air cylinder is 5x9, and the steam cylinder is 6½x9. It takes only three-fourths of a cord of green oak mixed with manzanita to run the compressor twenty-four hours, although the drill is run eighteen hours out of the twenty-four, and in addition a 4½x2¾ pump, elevating water 140 feet high, constantly day and night. I have to use cold water in the boiler, as I have no way to heat it. Am working in the hardest kind of rock. Where they only made 20 to 25 feet a month, when drilling by hand and working three eight hour shifts, I have made 112 feet in twenty-eight days."

In a second communication, Mr. Barlow says: "I have used every other make of compressors and drills, but consider the Cummings double-pipe system far superior to any other system in every respect. It makes no difference at all with the running of the drills whether they are close to the compressor or not. In the Grey Eagle Mine, in Placer County, my drills were 2,700 feet distant from the compressor. At Sunny Hill I have completed 500 lineal feet of tunnel in hard and tough rock with this system, and shall use the same machinery for another 300 feet for which I have taken a contract."

"The cost was only about half of the old style plants, besides the saving in foundation, etc.," says Mr. T. P. Chapman, of Denver, speaking of his experience in operating the Cummings system in drilling and pumping. "At Kingston, N. M., I ran a 2 inch pipe line from compressor to shaft, down shaft to 137 foot level, on which the air was taken out over 500 feet, and continuing on down the shaft to depth of 460 feet from that point the air was carried east over 500 feet and west over 250 feet, running three drills at the same time. My longest connection was about 1,000 feet. For economy in cost of operating, I would consider it good business judgment to remove any plant already installed and install the Cummings."

A report concerning a series of comparative tests made on a pump, both with the one-pipe and the two-pipe processes, was also furnished, but it is not reproduced here, as its conclusions were already given in Mr. Richard's last article in "Modern Machinery." Mr. Cummings claims that this system is particularly well adapted to machinery working at full pressure, and does not contemplate using the air expansively. This contention is fully supported by the preceding figures. He also admits that more numerous and elaborate tests than those on record are desirable, and namely, some comparative trials on rock drills, in both systems, the following testimonials merely stating that rock drills on the two-pipe process did satisfactory work, but without telling what these machines might have done with the other process. Circumstances did not permit, so far, to operate on a sufficiently large scale, to determine whether, on the long run, the alleged defects inherent to the system are of such a nature as to compensate its advantages.

Mr. Cummings admits the sluggishness of air flowing against a back pressure, and also the necessity of overcoming the inertia of the return column of air; in this respect, he expressed the idea that after the completion of a stroke of the motor's piston, the sudden expansion of the cylinderful of air at working pressure helps to overcome that inertia.

This contention, in the writer's opinion, is not evident; he believes that the actual facts were very ably presented by Mr. Leicester Allen, in a contribution to the "American Machinist," namely, that the expansive work developed by the live air escaping in the return pipe is first converted into velocity, and almost instantaneously transformed by friction into heat imparted to the escaping air; and that a very small fraction, if any, of this expansive work, is actually used in overcoming the inertia of the return column of air, whose mass, moreover, and unless the compressor and the motor are close to each other, is considerable, as compared with the mass of the escaping air.

The writer desires to say that he was very favorably impressed with the earnestness of Mr. Cummings' views of the subject, and he thinks that within the field of action indicated by that gentleman, more extensive tests of the two-pipe system would be conducive to useful and satisfactory conclusions.

A. E. CHODZKO

3 California street, San Francisco, Nov. 12.

Doubling the Efficiency of Compressed Air.

BY FRANK RICHARDS.

In the volume of the AMERICAN MACHINIST for the year 1898, at page 219, will be found an article with the above title in which I showed the promise of largely increased economy in compressed air power transmission by the use of the dense air or return pipe system. In this system the same air is used over and over (a small auxiliary compressor making up for the leakage) and the pressure in the exhaust or return pipe is maintained far above the normal atmospheric pressure. In the example which I considered in the article referred to, the assumed working pressure was 200 pounds gage, and the exhaust pressure 100 pounds. Theoretically the system promised a great increase in efficiency, and this has since been realized in practice. Several applications of the system have been made upon the Pacific coast and all have shown most satisfactory results.

I am enabled to present here some data from an installation of this system at the Bisbee West Mine, near Bisbee, Arizona. The plant consists of the following: One Ingersoll-Sergeant Straight-Line air compressor, steam cylinder 16 inches diameter, air cylinder 12¼ inches, stroke 18 inches; one Cameron station pump, air cylinder 16¼ inches diameter, plunger 6½ inches diameter, stroke 18 inches. A 6x6x6-inch auxiliary air compressor is connected to the low-pressure pipe line to supply leakage in pipe line, thus keeping the pressure constant at any desired point. The duration of the test here recorded was 70 minutes:

AVERAGE TEMPERATURES.

	degrees F.
Inlet to compressor	69½
Receiver	162
Inlet to pump	88.8
Discharge from pump	51.6
Station	72
Engine-room	93

GAGE PRESSURES.

	pounds.
Inlet to compressor	83
Discharge from compressor	158.3
Inlet to pump	152.6
Discharge from pump	90.9

MEAN EFFECTIVE PRESSURES IN CYLINDERS.

	pounds.
Compressor	65.6
Pump	61.7

SPEED.

	R. P. M.
Compressor	56.35
Pump	19.9
Auxiliary compressor	98.7

WORK.

Total indicated foot-pounds in air cylinder of compressor per minute	1,268,400
Total indicated foot-pounds in air cylinder of pump, per minute	751,500

The power thus indicated at the pump is 59 per cent. of that of the compressor cylinder, while with the usual compressed air practice in driving a steam pump in this way it is rarely that more than 25 per cent. is realized.

This plant is criticised for its deficiencies by the operator of another dense air plant, where a rock drill is operated instead of a pump. The drill is 1,800 feet from the compressor, and there are two lines of 2-inch pipe 1,300 feet long and two lines of 1½-inch pipe 500 feet long. The working pressure is from 215 to 220 pounds, and the return pressure 95 to 100 pounds. Notwithstanding that it is much more difficult to keep the piping around a drill tight than it is around a pump, the auxiliary compressor to make up the leakage is not run more than a quarter of the time. The compressor in this case, with air cylinder 5 inches diameter by 9 inches stroke, runs 90 to 95 turns per minute, and besides driving the 2¾-inch rock drill in "the hardest rock I have ever seen" it also runs a 5x6-inch slide valve engine eight hours a day to drive a blower, and pumps 8,000 gallons of water 200 feet high. The joints of the pipes in this case were made with shellac, which the engineer claims accounts for the small leakage. He also suggests that the pipes at the Bisbee West mine are too small, being 2½ and 3 inches instead of 3 and 4 inches, the latter of each of course being for the exhaust.

I regret the crudeness of the information here given, but it still seems clearly to show that, both theoretically and practically, there is much to be saved by the adoption of this system wherever there is any permanency of installation. The above data came from Mr. F. H. Wheelan, president of the Pneumatic Power Company, 224 California street, San Francisco, Cal.

INCREASING THE EFFICIENCY OF COMPRESSED AIR AT THE BISBEE WEST MINE

It is possible to prove theoretically that if the exhaust air from an air driven pump, or drill, be carried back to the compressor and used over again, there results a very great increase in the efficiency of the plant. In a recent issue of the "American Machinist," Mr. Frank Richards gives some data of an actual test secured from Mr. F. H. Wheelan, President Pneumatic Power Co., 224 California St., San Francisco. The data follow:

I am enabled to present here some data from an installation of this system at the Bisbee West Mine, near Bisbee, Arizona. The plant consists of the following: One Ingersoll-Sergeant Straight-Line air compressor, steam cylinder 16 ins. diameter, air cylinder 12¼ ins., stroke 18 ins.; one Cameron station pump, air cylinder 16¼ ins. diameter, plunger 6½ ins. diameter, stroke 18 ins. A 6 x 6 x 6-in. auxiliary air compressor is connected to the low-pressure pipe line to supply leakage in pipe line, thus keeping the pressure constant at any desired point. The duration of the test here recorded was 70 minutes.

Average Temperatures.

	Degrees F.
Inlet to compressor	69¼
Receiver	162
Inlet to pump	88.8
Discharge from pump	51.6
Station	72
Engine-room	93

Gage Pressures.

	Pounds.
Inlet to compressor	83
Discharge from compressor	158.3
Inlet to pump	152.6
Discharge from pump	90.9

Mean Effective Pressures in Cylinders.

	Pounds.
Compressor	65.6
Pump	61.7

Speed.

	R. P. M.
Compressor	56.85
Pump	19.9
Auxiliary compressor	98.7

Work.

Total indicated foot-pounds in air cylinder of compressor per minute	1,203,400
Total indicated foot-pounds in air cylinder of pump, per minute	751,500

The power thus indicated at the pump is 59% of that of the compressor cylinder, while with the usual compressed air practice in driving a steam pump in this way it is rarely that more than 25% is realized.

This plant is criticised for its deficiencies by the operator of another dense air plant, where a rock drill is operated instead of a pump. The drill is 1,800 ft. from the compressor, and there are two lines of 2-in. pipe 1,300 ft. long and two lines of 1½-in. pipe 500 ft. long. The working pressure is from 215 to 220 lbs., and the return pressure 95 to 100 lbs. Notwithstanding that it is much more difficult to keep the piping around a drill tight than it is around a pump, the auxiliary compressor to make up the leakage is not run more than a quarter of the time. The compressor in this case, with air cylinder 5 ins. diameter by 9 ins. stroke, runs 90 to 95 turns per min., and besides driving the 2¾-in. rock drill in "the hardest rock I have ever seen" it also runs a 5 x 6-in. slide valve engine eight hours a day to drive a blower, and pumps 8,000 gallons of water 200 ft. high. The joints of the pipes in this case were made with shellac, which the engineer claims accounts for the small leakage. He also suggests that the pipes at the Bisbee West Mine are too small, being 2½ and 3 ins. instead of 3 and 4 ins., the latter of each of course being for the exhaust.

The Journal of The Transvaal Institute of Mechanical Engineers.

1905? p.4-16
see also article by Behr in the Transactions of the Mechanical Engineers' Association of the Witwatersrand, 1905

THE RETURN PIPE SYSTEM OF COMPRESSED AIR POWER TRANSMISSION.

By Mr. H. C. Behr.

REPLY TO DISCUSSION.

The bringing forward of what might almost be termed a new method of power-transmission has naturally called forth comparisons with transmission by electrical means. The latter is generally the most applicable and most convenient power, but conditions are often encountered where other methods are more suitable than electrical.

It was such a set of conditions for a given case of mine drainage, which, about one and a half years ago, made it necessary for the writer to compare the merits of electricity with other methods, and in this connection to investigate the possibilities of the Dense Air System very fully. The general results of these investigations were given in the original paper.

The design of a plant for the case in question was worked out in detail, and estimates of cost prepared. It was intimated, in the course of the discussion, that the details of this estimate should have been given instead of only the approximate total, as in the original paper. The following is the itemised list of costs:—

Detailed Cost of Dense Air Pumping Plant.*

1. Boiler Plant.

(Two-thirds used for general purposes and one-third for pumping plant).

1. 6–220-h.p. Boilers erected (34½ lbs. water from and at 212 deg).	£7,760	
2. Flues	258	
3. Economisers (320 tubes erected)	2,650	
4. Induced Draught Plant, with Smoke Stock 100 ft. high	1,646	
5. Ash Haulage	1,290	
6. Steam Piping	303	
7. Feed and blow-off service, feed pumps, Hotwell, etc.	994	
8. Boiler House, Coal Bunkers, approaches, Pump House, etc.	3,370	
9. Electric Light	50	
10. Fire Service	150	
	£18,471	
Main Plant portion	12,314	
Pumping Plant portion		£6,157
11. Superheater with piping erected		1,107
		£7,264

* This detailed estimate of the cost of the dense air plant was prepared by Mr. G. H. Thurston, who, at the writer's suggestion, intended to present it with further remarks of his own. Unfortunately Mr. Thurston's temporary absence has prevented this, and the writer is therefore obliged to include it with his own remarks and estimates.

2. Compressor Plant.

12. Compressor with Condenser and Exhaust Piping, erected	£9,554	
13. Steam Piping, Separator, Covering, erected ...	282	
14. Circulating Service, Cooling Pond, Cooling Tower ...	1,090	
15. Air Piping, Receiver, etc. ...	2,773	
		£13,699

3. Pumping Plant.

16. Two Duplex Differential Pumping Engines with Foundations, Pipe Connections and Valves, erected	£6,837	
17. Pump Station Excavation, Floor and Travelling Crane	2,385	
18. Suction from Sump ...	260	
19. Pipe Column and Supports	1,857	
		£11,339

Summary.

1. Boiler Plant and Superheater ...	£7,264
2. Compressor Plant	13,699
3. Pumping Plant	11,339
	£32,302
10% contingencies, say	3,230
	£35,532
4% Engineer's Fees, say	1,421
Grand Total	£36,953

A few notes and comments might be made on the plant forming the basis of the foregoing estimate.

It will be seen that the estimate is based upon a first-class installation, including all the various auxiliary machinery and appliances that help to secure economy of operation in a modern power plant. The boilers considered are of the best British manufacture, and their cost is comparatively high.

The boiler plant forms an extension of an existing plant so as to reduce the amount of spares, and is proportioned for a steam consumption of 15 lbs. per i.h.p. hour. With independent boiler plant entirely separated from an existing plant, three boilers would be needed, so as to have one as a spare. The total cost of boiler plant would then be increased by about £2,000, so that the total cost of plant will come to £38,953.

Should the steam consumption amount to only 12 lbs. per i.h.p. hour, which would easily be the case with superheated steam under constant load, as in the present instance, then the size of boiler could be correspondingly reduced, and the cost of plant would remain practically as given in the above estimate.

Connected with an existing boiler plant and using at the same time only 12 lbs. of steam per i.h.p. hour, the cost of the plant would be reduced to about £35,500. If, instead of an independently fired superheater, one of the type built in with the boilers and forming an integral part thereof were used, a reduction would also result on this account.

Equipped with direct acting pumps having a compensating device, so as to enable using the air expansively as in the rotative type, the dense air plant would, with boilers connected to existing plant and using 12 lbs. of steam per i.h.p. hour, cost only about £33,500.

In order to compare the cost of the foregoing dense air plant with an electric installation on the same basis the writer has prepared the following estimate: —

Cost of Electric Pumping Plant.

1. Boiler Plant.

1. Boiler Plant with Flues, Economisers, Induced Draught Plant, 100 ft Stack, Coal Bunkers, Steam Piping, Superheater, Building, Ash Haulage, etc (under similar conditions as in the case of a Dense Air Plant) ...	£6,400	
		£6,400

2. Generating Plant.

2. 500 kw. 3,300 volt 60 cycle Turbo-Generator, Foundations and Switchboard erected ...	4,600	
3. Steam-driven Exciter Set with Foundations and Switchboard erected	450	
4. Steam Piping and Covering, Separator and Traps ...	250	
5. Condensing Plant, with electrically driven Air and Circulating Pumps, Transformers, Circulating Pipe, Exhaust Pipe, Excavations and Foundations	2,400	
6. Cooling Pond and Tower ...	900	
7. Building over Generator and Condensing Plant	800	
		£9,400

3. Pumping Plant.

8. 2 Motor-driven Geared Pumps, (parts made small enough to go down 4 ft. x 6 ft. compartment), Foundations, Pipe Connections and Valves, erected	7,000	
9. Transformers, erected ...	1,200	
10. Switchboards ,, ...	200	
11. Suction from Sump ...	260	
12. Cable in Shaft	300	
13. Ventilator for Station ...	50	
14. Cutting Station with Floor and Crane over Pumps ...	2,385	
15. Pipe Columns, installed ...	1,857	
		£13,252

Summary.

1. Boiler Plant	6,400
2. Generating Plant	9,400
3. Pumping Plant, Pipe Column and Cable	13,252
	£29,052
10 per cent. contingencies, say ...	2,905
	£31,957
4 ,, ,, Engineer's fees, say ...	1,278
Grand Total	£33,235

The well-designed electric pumping plant recommended by Mr. Peirce would cost for underground plant alone at least £4,000 more than the above pumping plant on which the writer's estimate is based. In addition, the extra capacity required by Mr. Peirce for eleven hours per day would demand corresponding extra capacity of generating and boiler plant, amounting to, perhaps, £1,500. Add 10 per cent. contingencies, and 4 per cent. engineer's fees, and the total increase of cost runs up to £6,300, making the total cost of the electric pumping plant under these conditions £39,500, or higher than that for the dense air plant.

The total cost of the underground part of the electric pumping plant could be reduced if centrifugal pumps were employed for the single lift of 1,200 ft. The efficiency would then, however, be lower, and the generating plant would have to be enlarged, though, perhaps, not to such an extent that its increased cost would equal the reduction due to the use of centrifugal pumps. The cost of the full installation would still be very much in excess of what Mr. McCann designates a liberal estimate, and which, for comparison, is repeated here :—

(*a*) Boiler Plant	... £3,000
(*b*) Engine and Generator ...	... 4,700
(*c*) Cable	... 300
(*d*) Motors and Transformers	... 1,500
(*e*) Pump	... 3,500
(*f*) Cutting Station (a guess)	... 5,000
(*g*) Building	... 1,000
(*h*) Pipe Line	... 2,000
Total	£21,000

Item *a* of the above is comparable with item 1 of the writer's estimate, and it is evident cannot have been based on a complete plant.

Item *b* must be compared with the sums of items 2, 3, 4, 5 and 6 of the writer's estimate for electric plant, which amount to £8,600, and barely covers what is actually needed.

Item *d* includes the pump motor, which, in the writer's estimate, is taken together with the pumps. The transformers alone would cost at least £1,000, without any spares, and a motor to drive a pump delivering 515 h.p. in water lifted can certainly not be obtained for the remaining £500. Items *d* and *e* taken together, and compared with the corresponding sum of items 8, 9 and 10 of the writer's estimate, give £5,000 as against £8,400. Mr. McCann does not enlighten us as to what type of pump item *e* represents.

Item *f* is qualified by the note that it is a guess. In view of experience, the guess must be considered certainly very wide of the mark, being more than twice the needed amount.

In the writer's scheme for both the dense air and the electric plants no spares were included except in the case of the boilers. In considering the surface plants alone, both are in about the same condition to meet a moderate increase of load without additional reserve plant.

In the case of the underground plant, however, the dense air installation is in a better position, as it can be simply speeded up to immediately meet an increase in the amount of water, while the electric plant would require a corresponding reserve pump and motor held in readiness to meet the increase at once. The capacity of the electric plant, it is true, could be increased by insertion of larger plungers, or by a change of gears giving a higher pump speed, if provision for such arrangements were made. But besides increasing the cost, either of these two plans would involve the stoppage of part of the plant for a considerable time. The former method, by providing a spare pump, as suggested by Mr. Peirce, would be, no doubt, the better plan, and its cost would probably be little greater than the provision for change of plungers or gearing. At any rate it seems necessary in the electric plant to provide for some reserve, which would not be needed with the dense air plant for, say, a 20 per cent. emergency increase. The cost of the electric plant should, therefore, be higher than the figures given by the writer.

Concerning the operation under water which the writer mentioned as an advantage of the dense air plant, Mr Heather maintains that this can also be accomplished by a suitably arranged electric plant, and he states that it has been done long ago. He also mentions that tramway motors are arranged to work under water. The writer does not doubt that an electric motor might be so enclosed that it could work under a moderate submersion for a short time, but, for heavy pressure and continuous operation, means would have to be provided to remove the water leaking in at the stuffing boxes around revolving shafts. This might be done either by a small pump or by air pressure within the casing exceeding the outer hydraulic pressure. The high speed bearings of the motor would probably soon give trouble, as they would be without attention and lubrication for considerable time unless special arrangements for oiling from the surface were devised. With the much slower speed of the dense air pumps lubrication is not so serious a matter, especially if the pumps are direct acting. In the case of actual practice at the Bisbee West Copper Mine, mentioned by the writer in the original paper, there seems to have been no trouble whatever, even under conditions of extreme neglect.

Mr. Epton and Mr. Cooke remark that there is no need to provide for submerged operation of the pumps. This may be true for most cases, but there are undoubtedly conditions and times when such a provision is most desirable. A well-known case on the Rand is the Knights Deep, which has been repeatedly in danger of becoming flooded through sudden increase of water. If in such a case as this the additional security can be obtained at the same, or even a somewhat higher, cost of plant or operating expense, it would seem that the means available which afford such extra security should be seriously considered in selecting the type of pumping plant.

As to operating costs of the dense air and of the electric plants :—In order to properly compare these it will, in the first instance, be necessary to present more details than given in the original paper. The costs for the dense air plant there given were based upon the following detailed estimate :—

Detailed cost of operation (on the Rand) of a Cumming's Dense Air Plant for 2,000,000 gallons per 24 hours from 1,200 ft. depth. (Boiler plant forms extension of an existing plant.)

I.—Surface Plant.

1. Labour.

$\frac{1}{2}$ of 3 engine drivers at £1 per 8 hour shift	£1	10	0
3 engine-room boys at 3s. per 8 hour shift	0	9	0
$\frac{1}{3}$ of 3 white firemen at 15s. per 8 hour shift	0	15	0
$\frac{1}{3}$ of 9 fire boys at 6s. per 8 hour shift (including food) ...	0	18	0
$\frac{1}{3}$ of 9 ash boys at 4s. 6d. per 8 hour shift (including food) ...	0	13	6
	£4	5	6

2. Coal.

(*a*). For 15 lbs. steam per i.h.p. hour, evaporation 5 to 1, about 30 tons per day at 13s.	£19	10	0
(*b*). For 15 lbs. per i.h.p. hour, evaporation 6 to 1, about 25 tons per day at 13s. ...	16	5	0
(*c*). For 12 lbs. per i.h.p. hour, evaporation 6 to 1, about 20 tons per day at 13s. ...	13	0	0

3. Stores.

2 gals. engine oil	£0	3	6
1 gal. cylinder oil	0	6	0
5 lbs. of waste	0	1	3
Sundry stores	0	10	0
	£1	0	9

4. Other Items.

Maintenance on engine and compressor	£0	6	0
Maintenance on boilers ...	1	0	0
Electric light	0	5	0
Supervision, manager and engineer, and office charges ...	1	0	0
	£2	11	0

Total Operating Cost of Surface Plant.

	Per Day.			Per Month.		
With coal as under *a*	£27	7	3	£820	17	6
,, ,, ,, *b*	24	2	3	723	7	6
,, ,, ,, *c*	20	17	3	625	17	6

II.—Underground Plant.

1. Labour.

3 white attendants at 20s. per shift	£3	0	0
3 native helpers at 4s. ,,	0	12	0
	£3	12	0

2. Stores.

2 gals. engine oil	£0	3	6
1 quart cylinder oil (drill oil) ...	0	1	6
1 lb. grease	0	1	0
4 lbs. waste	0	1	0
Packing (30 lbs. per month) ...	0	6	0
Sundry stores	0	3	0
	£0	16	0

3. Other Items.

Maintenance	£1	2	0
Electric light (10 16 c.p. lamps) ...	0	5	0
Supervision (Manager and Chief Engineer) and office charges ...	0	18	0
	£2	5	0

Total Operating Cost of Underground Plant.

Per day ... £6 13 0 Per month £199 10 0

Total Operating Cost of Entire Plant per Month.

	Surface.	Underground.	Interest and Depreciation.	Total.
With coal as under				
a.	£820 17 6	£199 10 0	£370 0 0	£1,390 7 6
b.	723 7 6	199 10 0	370 0 0	1,292 17 6
c.	625 17 6	199 10 0	370 0 0	1,195 7 6

There are some other items of expenditure which, strictly speaking, should be added, such as insurance, hospital charges, etc., which, however, the writer could not accurately determine, and which would bring up the total cost above the figures given. But, as these items are also omitted in the estimate for operating cost of electric plant, given further on, for which they can be assumed as practically identical, any difference in the operating costs for two plants would not be affected.

The item "fuel" has been given under three different conditions in the foregoing. The most unfavourable of these was taken for the estimate in the original paper. It must be remembered, however, that the plant given in the estimate of first cost includes the most perfect steam installation, and, unlike the ordinary drill compressor plant, operates under the most favourable conditions of constant speed and constant load, as, for example, the engine driving a mill. The 1,000-h.p. engine driving the Simmer and Jack mill, with a vacuum of only 13 inches, uses only 14·5 lbs. of saturated steam per i.h.p. hour. It is, therefore, only fair to assume that with superheated steam, as in the pumping plant under consideration and with a better vacuum, not over 12 lbs. of steam per i.h.p. hour should be consumed by an 850-h.p. engine running under constant conditions. As the boiler plant would also operate under constant conditions, an evaporation of 6 to 1 should be easily obtained with coal of 10,000 B.T.U. calorific value.

The operating cost of the electric plant will under similar conditions be about the same for labour, stores, maintenance, light, supervision, and office charges. The item "fuel" might possibly be lower. With a steam turbine using steam of 150 lbs. gauge superheated 150 degrees, and with a high vacuum of only 1 lb. back pressure, a consumption of 18 lbs. of steam per k.w. hour is said to be obtainable. With an evaporation of 6 to 1, 3 lbs. of coal per k.w. hour would be required. With 500 k.w.'s at the surface 18 tons of coal, costing 13s. per ton, or £11 14s. per day would be required. The operating cost then works out as follows:—

Labour ...	£7	18	7	About same as for Dense Air.
Oil and Stores ...	1	16	0	
Maintenance, Supervision, Light and Office Charges ...	4	15	0	
Fuel	11	14	0	
	£26	4	7	per day.
	£786	17	6	per month.
Interest and Depreciation at 12 per cent. on £33,235 ...	£332	7	0	

Total operating cost £1,119 4 6 per month, as against the £938 10s. 0d. of Mr. McCann's estimate.

With regard to the steam consumption of the turbo-generator on which the writer's estimate is based, it should be noted that the 18 lbs. per k.w. hour are only obtainable with the high vacuum of 1 lb. back pressure. Such a vacuum has been found difficult of attainment on the Rand. It is also to be remembered that the estimate for condensing plant was the same for the dense air plant as for the steam turbine, which would not represent the actual conditions on the Rand for a steam consumption of 18 lbs.

per k.w. hour. For this case the cost of condensing plant, cooling tower and circulating service would be higher than was given in the estimate, though the size and cost of the boiler plant would be less.

First class slow speed reciprocating sets do not require so high a vacuum as steam turbines in order to give the best results, but with them the total cost of plant, even when considering the reduction in cost of boilers and condensing plant, becomes considerably higher than that of a turbine plant.

If the current for operating the pumping plant be taken from another concern operating a large central station, the amount allowable for this must not be more than the difference between the total operating cost of an individual plant at the mine and that of the underground plant alone.

The operating cost of the underground plant, whether run by electricity or dense air, will be about as given previously for the latter kind of plant. Taking the general items as follows:—

Labour	£3	12	0
Stores	0	16	0
Maintenance	1	2	0
Electric Light	0	5	0
Supervision and Office Charges	0	18	0
Total Daily Expenditure ...	£6	13	0
Total Monthly Expenditure	£199	10	0
Interest and Depreciation on Underground Plant at 12 per cent. on £13,252	132	10	0
Total cost per month ...	£332	0	0

Taking the highest and lowest total cost of operation as given for the dense air plant, the difference which could be allowed to be paid for current derived from a central power company, in which the mining company is not interested, would be as follows:—

	Highest.	Lowest.
Total monthly cost of operating Dense Air Pumping Plant	£1,390	£1,195
Monthly operating cost of Underground Plant	332	332
Amount that could be paid to Power Company	£1,058	£863

The largest value corresponds to a price of ·705d. and the smaller to ·575d. per unit.

If the cost of operating an electric pumping plant at the mine could be brought down to £1,119, assumed by the writer as a possible minimum for the conditions, then the price that could be paid to an electric power company would be £786 per month, or ·52d. per unit.

It should be understood that these figures must include, besides the expenditure returns, interest and redemption of capital on the proportion of plant required at central station, interest and redemption on the cost of power line, and also a percentage for profit on the capital invested in central station capacity and line.

As a matter of fact none of the central stations on the Rand have so far been able to supply current at a figure anywhere near to those named in the foregoing, but it is only fair to add that their price must include insurance, hospital charges, etc., not taken account of by the writer for the individual plants. This would slightly raise the figures.

One must concede that by electric transmission very high efficiencies have been obtained, but few engineers, electrical as well as others, would be inclined to accept the statement that the figure of 78·6 per cent. would represent every day practice as claimed by Mr. McCann.

The table on page 10 was compiled by Mr. A. Spier of the A. E. G. Electric Company from an article in the *Zeitschrift des Vereins deutscher Ingenieure*, No. 42 to 53, 1904.

The highest total efficiency given under the heading of pumps is less than 70 per cent. A total efficiency generally assumed attainable in practice with a first class electric pumping plant is between 66 and 68 per cent.

The efficiency on which the estimates of capital expenditure and operating cost for the dense air plant were based by the writer is about 60 per cent. This was for a plant by itself, not connected with the air drill plant of the mine. In case all the air exhausted by the pump could be utilised for the drills, the efficiency, as given by the writer in the original paper, would reach nearly 70 per cent., or about the same as in the case of the most efficient electric plant in the table.

In comparing winder efficiencies the 49·6 per cent., given by the writer, contains a factor of 78 per cent. for efficiency of winder mechanism, shaft friction, etc. Up to the winder the dense air efficiency, or efficiency of the process, was 62·88 per cent. The 78 per cent. would also have to enter as a factor into any electric winding plant operating under the same conditions.

The writer recently had occasion to call for tenders for an electric winding plant of large capacity to haul from an incline shaft, and electrical engineers, who had large experience with electric winding plants, stated that the total efficiency of plant would be in the neighbourhood of 45 per cent., *i.e.*, the work of raising the useful load would be that percentage of the indicated work of the engine. The speed of the A.C. motor of the winder was in this case to be controlled by liquid resistance, so that there was some loss during the acceleration period. With the Ilgner system, under favourable conditions, better results are claimed to be obtained, *i.e.*, with trips in quick succession, a condition more easily carried out in vertical shafts than in inclines, where the ore has to be gathered from a large number of chutes, and delays of many minutes are frequent. Under such conditions as these the power required to constantly run the motor-generator with its heavy fly-wheel may be a considerable addition to the work required, and would materially reduce the efficiency of the plant if long stops are of frequent occurrence. In the instance of the winders required by the writer, he was advised by electrical engineers of experience, who carefully examined the conditions, that in the case under consideration the resistance control of the winders would ensure the cheapest operation. The air resistance to the motion of the large Ilgner fly-wheel running at high velocity, together with the motor-generator losses, amounted, according to the engineer of one of the electrical companies, to 40 h.p. continuously. As the installation was to be placed at a depth of 2,000 ft. from the surface, the fly-wheel would have had to be made in parts small enough to go down the shaft, which would have been a somewhat expensive undertaking, and the cost of the entire installation would have been very much higher than that of the alternating current plant with resistance control. The cost of operation would therefore have been higher, not only on account of operating with longer intervals between trips, but also on account of the higher interest and amortization charge.

A dense air plant would have cost no more than the alternating current installation, and the efficiency would probably have been a little higher. But additional electric plant was, in this case, required for other purposes, so that a large plant became available to take up and reduce the peaks of the fluctuating load caused by the winder, and the electric method was therefore adopted.

There is one advantage with the dense air winder, in that like a steam winder, it can run with full efficiency at any speed up to a considerable percentage above the normal. The longer intervals between trips will also not affect the economy until the intervals become so long, and the power demand so reduced, that the compressor has to operate at very uneconomical speed. By arranging that for the slowest speed of the compressor its capacity can be still further reduced by re expulsion of part of the air taken in during the suction stroke, so that a smaller volume of air is compressed per stroke, it should be possible with a dense air system to work economically within the widest usual range of output and length of interval between trips.

In regard to efficiency of electric winders, the writer would call attention to articles in the *Iron and Coal Trade Review* of June 2nd, and in *Page's Weekly* of June 9th, 1905, both of which papers contain abstracts of a supplement to a paper on the "Electrical Driving of Winding Gears," by F. Hird. The paper was read before the Institution of Mining Engineers during the meeting of June 1st, 1905. The following is a copy of the abstract given in *Page's Weekly*:—

ELECTRIC DRIVING OF WINDING GEARS.

At the general meeting held in London in July, 1903, Mr. F. Hird read a paper on the above subject, and drew attention to the then newly-developed Siemens-Ilgner system. It was at that time impossible to give any actual figures as to results, as no installations were then at work. Actual tests are now available.

The plant on which the following tests were taken is installed at Friedrichshall, at the König-Wilhelm II. shaft, and comprises a winding gear, consisting of a double-cylindrical drum-winder, driven by one direct-coupled motor. The starting apparatus consists of a variable-voltage continuous-current generator, driven by a three-phase motor of 2,000 volts, 50 periods, the two being coupled to a fly-wheel weighing 8·8 tons and having a diameter of 8 ft. 3 in. The speed of this set is 600 revolutions per minute at an absolute maximum, and falls to 15 per cent. below this speed, when the greatest demand for power is made. The power delivered by the variable-voltage generator reaches a maximum of 305 h.p., which occurs at the end of the acceleration period. The power then falls to 215 h.p., when the constant speed of the cage is attained, and continues to fall to 95 h.p. A separate small motor-generator gives the necessary exciting-current for the winding-motor and for the motor-generator. The shaft has a depth of 617 ft., and the contract specified that two full trucks, containing 1,543 lbs. each, were to be raised at each lift, and that 55 lifts per hour must be made. At the first test, 55 lifts per hour of two full trucks each were made, and the energy consumption was found to be 1·55 kw.-hours per lift. On a second test, 63 lifts per hour were attained, and the consumption of energy was the same as before; 1·55 kw.-hours per lift. During this test a maximum speed of 23 ft. (7 meters) per sec. was attained, and the time of one lift was 36 secs. The average load in the trucks was found to be 1,488 lbs. (675 kg.) The net useful work in each lift (2 tubs x 675 kg. x 188 meters) equals 253,800 kgmm., or 1,836,000 ft. lbs., viz., 0·927 h.p.-hour, equal to 0·69 kw.-hour. The electrical energy consumed was 1·55 kw.-hours, giving a total efficiency of (100 x 0·69 ÷ 1·55 or) 44·5 per cent. The load on the generating plant was found fairly steady, and the fly-wheel of the motor-generator was apparently ample for its work, the speed variation during working being no more than 7 per cent. The ease of manipulation left nothing to be desired; any speed from dead-slow to full-speed being easily obtained, and steadily maintained.

The efficiency of 44·5 per cent. given in the article, it will be observed, is based upon the k.w. output of the generator. In order to compare it with the efficiency of the Dense Air Winder in the example assumed by the writer, in which case the efficiency is referred back to the i.h.p. of the compressor-engine, the 44·5 per cent. must be multiplied by the combined mechanical and electrical efficiency of the generating plant. This is probably about 86 per cent. and the total efficiency of this electric plant, even when running with trips in quick succession, as stated in the paper, is then only 38·3 per cent. With long intervals between the trips, the efficiency would be considerably less.

However, the writer imagines that the case cited cannot be that of a representative first-class plant. Better efficiency should be obtained with a well arranged and intelligently operated installation. But the result shews that an electric plant can give just as bad efficiency as a poorly designed and negligently operated compressed air plant.

Mr. Schweder and Mr. Whittome suggested that it might be better not to connect the rock drill plant with the exhaust of the high pressure air, but to provide for use of the exhausted air in a secondary machine, the air being first warmed by transmission of heat from the air in the mine. The writer sees no reason why this could not be done, and, perhaps, a somewhat better efficiency obtained. The machinery required, however, would be much more complex and costly, while the re-heating arrangements, operating as they would between narrow limits of temperature, would require very large surfaces, and would, no doubt, be very costly. Interest and redemption charges and extra attendance would increase the operating cost probably more than a possible increase in efficiency might reduce it. One of the main advantages of the dense air system is the simplicity of its compressor and motor plant.

Mr. Vaughan asked if any experiments had been made to utilise the heat in the air of the mine for the purpose suggested by Mr. Whittome.

In the only attempt which the writer knows of, the air of the mine was drawn into and caused to flow around a jacket surrounding the cylinder of small engines, finally escaping through an annular nozzle around the exhaust. The device did not, nor could it, give any appreciably greater efficiency, but it prevented the exhaust from freezing up. This, however, can be more easily accomplished in practice by using a long exhaust pipe of some heat conducting material, which causes a certain amount of back pressure and transmits heat from the air of the mine to the exhaust air in the pipe.

Mr. Vaughan raises the point whether or not there is a decrease of pressure due to decrease of temperature in the air pipe leading from the compressor at the surface to the air engine underground.

It must be considered that for constant conditions the same weight of air per unit of time must be introduced at the inlet of the pipe which is withdrawn at the outlet. It is evident also that there could be no loss of pressure due to drop in temperature in the case of a pump, since a definite volume of air at a given pressure is required to operate the pump at a given speed, and if this volume is colder at one time than at another, a very small drop in pressure due to contraction of volume would at once influence the pressure regulating device at the surface and cause the compressor to speed up in order to supply the amount of air needed at a fixed pressure. The volume of air delivered by the compressor is greater than the volume consumed by the pump in the ratio of the temperatures at the two points. That is, if we assume no resistance due to the friction in the pipe, and if the latter were horizontal, so that the total weight of air would be distributed over its lower side instead of being supported by one end as in the case of the vertical pipe. In the latter case the air would be compressed by its own weight, and there would, therefore, be more

TESTS ON UNDERGROUND PUMPS

MADE BY A COMMITTEE OF THE "VEREIN DEUTSCHER INGENIEURE" AND THE "VEREIN FÜR DIE BERGBAULICHEN INTERESSEN IM OBERBERGAMTSBEZIRK DORTMUND."

(*Electrical figures checked by* PROF. GRÜBLER, *Dresden*).

MINE.	System.	Kind of Pumps.	Diam. mm. (in.)	Stroke. mm. (in.)	Speed. r.p.m.	Delivery. litre p.m. (gal p.m.)	Height. m. (ft.)	Efficient h.p. in water lifted.	Indicated h.p. of steam engine	Total efficiency.	Efficiency of steam engine.	Efficiency of generator.	Efficiency of cable.	Efficiency of motor.	Efficiency of pump.	Steam consumption per indicated h.p.	Steam consumption per effective h.p.
Victor Raussel ...	Steam	Two double-acting plunger pumps	243 (9 9/16)	1,300 (51 3/16)	51	11,500 (2,540)	505 (1,658)	1,300	1,460	89·05						9·84 kg. (21·7 lbs.)	11·02 kg. (24·3 lbs.)
Dannenbaum No. 2	Hydraulic Schwartz-kopff	Four-plunger	325 (12 13/16)	800 (31½)	16	4,585 (1,010)	510 (1,672)	524	796	65·8						6·91 (15·2)	10·54 (23·15)
Victor Raussel ...	Electric	Two centrifugals in series Sulzer			1030	7,350 (1,620)	502 (1,648)	816	1,387	58·8	88·5 (unit cond.)	94·14 (unit excit.)	99·25	94·4	75·47	6·834 (15·05)	11·07 (24·4)
Adolf von Hausemann	Electric	Two double-acting plunger pumps (Express Schleifm)	166 (6½)	500 (19 11/16)	122·5	5,080 (1,120)	442 (1,450)	502	721	69·63	88·86 (unit cond.)	89·85 (unit excit.)	98·72	91·97	96·05	6·7 (14·75)	9·525 (21)
Colonia Mansfeld ...	Electric	Two single-acting plunger (Riedler Empress)	248 (9¾)	350 (13¾)	149	4,625 (1,020)	435 (1,427)	For two pumps 875	1,278	68·47	89·74 (unit cond.)	93·7 (unit excit.)	98·83	90·65	91·77	4·73 (10·4)	6·91 (15·2)

Compiled from *Zeitschrift des Vereins deutscher Ingenieure*, No. 49-53, 1904.—20.5.05, SPIER.

pounds of air in the vertical than in the horizontal pipe. Without considering friction, the pressure at the upper and inlet end of the vertical would be the same as at the outlet of the horizontal. The ratio of volume of the entering and discharged air is, therefore, larger with the vertical than with the horizontal pipe. Of course, the friction in the pipe counteracts the effect of weight and again reduces the pressure at the discharge end.

The weight of the air in the vertical pipe, and, therefore, the increase of pressure at the lower end, due to weight of air column, will also depend upon the rate of cooling of the air, and this cooling will again depend somewhat upon the velocity of the air.

That the air under ordinary conditions loses its heat very rapidly is shown by the following table of observations made with one of the compound compressors at the Robinson Deep Mine.

Rev. per minute of Compressor.	TEMPERATURE FAH. Compressor Delivery.	Receiver.	Outside Air.
23·7	205·	127·0	61·
28·8	216·	127·5	61·
40·5	224·	130·5	61·
50·2	253·	130·5	61·
61·3	264·	135·0	61·

The pipe from the compressor to the receiver, along which pipe the drop in temperature occurred, was not over 30 ft. long. It is therefore reasonable to assume that the air would reach the temperature of the atmosphere not far below the collar of the shaft. It is true that high temperatures of air in pipes have sometimes been observed at considerable distances from the surface, but in such cases the heat must have been due to burning oil, or coal dust ignited by the high temperature of compression in compressors. Mr. Johnston mentions a temperature of 320 deg. for the air, but he does not say whether this temperature occurred near the surface or underground.

Mr. Epton questions the possibility of keeping the air mains tight and refers to the defective condition of the air distributing system in our mines, where air leakage is a well recognised and possibly an unadvoidable loss. The unadvoidable leakage, however, will generally be due only to that part of the distributing system which runs from the mains in the shaft to the drills, small winches, or other apparatus at a distance from the shaft, where connections are frequently made and broken, and where the system is in the charge of unskilled and probably careless hands. In the shaft itself and in main drives there should be no difficulty in securing an absolutely tight air main. The writer once installed and operated a compressed air haulage for the tunnel of a drift mine which was three miles into the hill when the plant was put in. The total length of pipe from the compressor station outside to the last charging station inside was about two and a half miles.

The charging pressure was 750 lbs. per sq. in., and the loss in pressure when standing for an hour, which was frequently the case, was inappreciable as long as the valves at the locomotive charging stations were tight. The pipeline was only 2 in. in diameter, and contained, therefore a comparatively small volume of air, so that even a slight leak would, under the heavy pressure, cause a considerable reduction of this in a short time. Neglected stop-valves and stuffing-boxes of moving rods are the places where leakage is most likely to occur, but if they are of proper design, such as can be secured in large pumping and compressor plants, there should be no trouble from leakage at pressures like 160 lbs. above the atmosphere. The experience with ammonia-compressors in refrigerating plants indicates what can be accomplished in this respect. As a case from practice under adverse conditions, the account quoted by the writer in his original paper of the Bisbee West Dense Air Pumping Plant might be referred to again. In this case the leakage, notwithstanding the most extraordinary neglect, evident from the letters of the manager and foreman, seems much less than one would expect under such circumstances. It was very probably due to neglect of the stuffing boxes.

Mr. Epton's general remark, that small electric winders are simpler to operate than steam hauling engines does not seem sufficiently definite. A small electric winder should be compared with a small steam winder, and the handling of the latter is certainly not complicated. Large electric winders, like those operated by steam, generally require steam or compressed air machinery to operate brakes and clutches. The control of throttle and reverse lever of a single cylinder engine, like that of a dense air winder, can also not be more complicated than the control and reversal of an electric winder. The electric systems, requiring conversion of current from alternating to direct, seem certainly more complex than a dense air plant.

There is, however, pertaining to a well-designed electric winder a possible point of superiority over winders operated by reciprocating engines in the uniform torque and strain on the rope at starting attainable with careful handling. With the reciprocating engine there is a periodic variation of the pull on the rope which is most marked when starting. On the other hand, as pointed out previously, the reciprocating winding engine can be run at very much higher speeds than the normal for which it is built, while with the electric winder the possible increase of speed is much more limited. The possibility of a considerable increase in speed is most desirable for a winder, especially if bailing becomes necessary.

The dense air pump, like the winder, besides being able to operate when submerged, which may, however, be only a small advantage in most cases, can, as pointed out previously, be forced so as to operate at a higher output; certainly a desirable feature in any case, but especially where a level in a mine is liable to be flooded.

Mr. Vaughan has given a very clear explanation of the writer's method of demonstration for efficiency of dense air machinery. In reference to Mr. Vaughan's ingenious introduction of the fictitious temperature T_x, in dealing with the ordinary system of compressed air operation, it only has to be borne in mind that while the efficiency is correctly arrived at by this conception, the temperature T_x does not represent actual conditions, and that both final volume v_2 and pressure p_2 must be arrived at from actual final temperature T_2.

The writer agrees entirely with Mr. Vaughan that a pressure ratio of 2 to 1 with exhaust to atmosphere, demanding, as it does, low working pressure, and therefore excessive volumes of air cylinders, would be productive of great loss in mechanical efficiency. In reciprocating engines friction seems to depend more on size of machinery than on pressure producing friction.

Mr. Schweder doubts the exceedingly high mechanical efficiency of 96·7 per cent. of the compressor quoted by the writer as given by the Boiler Inspection Society of Dortmund. It seems rational to question such an efficiency, and Mr. Schweder's explanation of how errors could easily enter into such observations is very clear, but his arguments apply more particularly where reliance is placed on a single set of indicator cards. The Dortmund tests may be wrong on the high side, but it would hardly seem reasonable that the error could be more than a few per cent. Several sets of indicator cards would no doubt have been taken, and if these had shown great variation of efficiencies, the result would surely not have been accepted. There would not be any necessity for continuous diagrams if the governor were disconnected and constant conditions for speed and discharge of air were maintained. Constant speed can be easily maintained under control of tachometer indications by regulation of air discharge from the receiver.

In *Mines and Minerals*, of May 5th, there is published a very complete test of a Nordberg compound compressor, which gives the very high average efficiency of 95·1 per cent. The test extended over eight hours. The results of hourly indicator diagrams are given in the following table, abbreviated from the more complete table in the publication referred to:—

Indicator Card.	Time Taken.	R.P.M.	Indicated Horse-powers. Both Steam Cyls.	Indicated Horse-powers. Both Air Cls.	Mechanical Efficiency.
1	A.M. 10.45	90	291·66	276·16	94·6
2	11.45	91	297·04	285·47	96·1
3	P.M. 12.45	90	288·67	275·44	95·4
4	1.45	90	295·82	282·45	95·47
5	2.45	87	279·30	269·94	96·64
6	3.45	91	301·01	281·96	93·67
7	4.45	91	307·93	288·46	93·67
8	5.45	88	283·74	270·60	95·37
Averages			293·15	278·81	95·1

The last column in the table was derived from the two preceding columns and added by the writer. The variation of efficiency is very small, and cannot always have erred in the same direction, so that the error in this case at least is much less than Mr. Schweder fears.

Mr. Schweder is of the opinion that re-heating would be an advantage with the Cumming's system. There will no doubt be increased efficiency of the process, but notwithstanding this and the somewhat smaller, and therefore less costly plant and air pipe line, the commercial efficiency will not be greater and may even be lower, while at the same time there is introduced the inconvenience and complication due to re-heating.

Take for example the case of the pumping plant in the writer's original paper. The temperature of the air entering the air cylinder of the pumping engine was taken the same as that of the station, *i.e.*, 80 deg. Fah. or 540 deg. absolute. Re-heating, so that the temperature would be 240 deg. Fah. or 700 deg. absolute in the cylinder of the air engine, would give an exhaust temperature of about 110 deg. Fah. or 30 deg. above that of the mine. With this amount of re-heating there would be an increase in volume and available work of 30 per cent. between the same limits of pressure. The 854 i.h.p. at the surface could be, therefore, reduced to $\frac{854}{1 \cdot 3} = 657$ i.h.p. The pump H.P. was 515 so that the total mechanical efficiency becomes $\frac{517}{657} =$ ·78·4 per cent. instead of 60·3 per cent. as in the case without re-heating.

With a steam consumption of 12 lbs. per i.h.p. hour and an evaporation of 6 to 1, including re-heating, which should be practically attainable with coal having a calorific value of 10,000 B.T.U. there would be used 2 lbs. of coal per i.h.p. hour, or 2 x 854 = 1708 lbs. per hour without reheating, and 2 x 657 = 1314 lbs. with re heating, a saving of 394 lbs. of coal per hour, or 4.72 tons per day, which, at 13/-, cost £3 1s. 4d. at the surface.

The coal required underground would be arrived at as follows: The weight of air required without re-heating was found to be 16·86 lbs. per second. With 30 per cent. increase of volume by re-heating the weight would be reduced to $\frac{16 \cdot 86}{1 \cdot 83} = 12 \cdot 96$, say, 13 lbs. per second, or 46,800 lbs. per hour. To raise the temperature of this amount of air from 80 deg. F. to 240 deg. F. requires the expenditure of (240 – 80) x 46,800 x ·2,377 = 1,779,897 B.T.U.

Assuming for the moment that the use of ordinary steam coal is permissible underground, the re-heated air will probably reach the cylinder space with not over one-half the heat units contained in the fuel, the loss being due to imperfect combustion and to radiation from the re-heater as well as from pipes and cylinder walls. With coal of the same calorific value (10,000 B.T.U.) as that used with the plant at the surface 5,000 B.T.U. would then be utilised and the coal required per hour would be $\frac{1,779,897}{5,000} = 356$ lbs., or 4·22 tons per day, costing at the surface £2 14s. 8d. or only 6/8 less than the cost of coal saved at the surface.

If the efficiency of the re-heating process were 70 per cent. instead of 50 per cent., the amount of coal used would be only 3·048 tons per day, costing £1 19s. 7d. at the surface, or £1 1s. 9d. less than the coal saved at the boilers. This saving would be easily wiped out by the extra cost of handling the coal underground, removal of ashes, etc.

Against the reduction in cost of surface plant and pipe line must be placed the cost of the re-heater, with coal bunkers, coal handling appliances, and the necessary underground extra excavation, which may cost just as much as the saving in cost of surface plant. The inconvenience of handling the coal and ashes in the shaft, together with the higher temperature at the pump station, would also count against any system requiring re-heating, unless a very large economy were effected in other directions, which, from the foregoing, does not seem reasonable with the dense air system.

The foregoing deductions were based on the assumption that ordinary coal could be used underground. This is probably not admissible, and the use of coke, the price of which on the Rand is many times that of coal, would be entirely out of the question, even though the quantity of coke used migth be from 25 to 30 per cent. less than with coal.

There is one remark of Mr. Schweder's which, though only somewhat casual, recalls a subject which the writer considers interesting and important. The operation of air drills by electrically driven compressors close to the drills, has been rather a favourite scheme with the writer for some years. In order to replace the ventilation with oil laden air sent through the drills from the compressor at the surface, larger volumes of fresh air could be supplied at low pressure, either in a direct manner by electrically driven fans, through pipes leading from the downcast shaft, or in an indirect manner by the removal of the foul air by means of such fans, which would then send it into the upcast shaft, thus inducing a flow of fresh air to replace the foul air removed. In order to reduce the required size of pipes, a number of fans could be inserted at intervals in the line of pipe, one fan taking in at its suction end the air delivered from the preceding fan. The writer described practically the same idea on page 384 of volume XI on Deep Winding, published by the Institution of Mining and Metallurgy.

It has been urged against this method of ventilation that its cost would be so great that the extra interest and redemption charges would exceed any gain by operating the drills in the manner suggested. The writer is not aware, however, that any figures have been brought forward in support of this objection. The question of ventilation is of importance in connection with the operation of rock drills, either directly or indirectly, by electrical means, as in the case brought forward, and reference to it here was thought, therefore, not to be out of place.

Notwithstanding the arguments against the small portable compressor on the ground of its not producing ventilation near the face, a number of compressors of this type are gradually being introduced in Europe and in the United States. Such compressors are even now an article of manufacture. The Reavell compressor manufactured in England is of this type. In the May number of *Mines and Minerals*, 1905, there is described and illustrated a portable electrically driven compressor, manufactured and placed on the market in the United States The writer is informed that one of the Reavell machines is to be brought out to the Rand for the purpose of making experiments with it.

In his original paper the writer stated that he did not consider the Cummings system well adapted to the operation of drills. In expressing this view he had in mind the applications as carried out in California with compressors at the surface and with the long double line of pipes through the drives and workings of the mine.

If, however, the compressor be placed close to the drills, the long lines of pipes under heavy pressure in drives and stopes, carelessly put together by unskilled hands, are done away with. There would then only remain the objection of the double hose connection to the drills, which small objection should count little in view of the other advantages of higher efficiency and greater

flexibility of plant. With the Cummings system thus applied there would be no great heat developed by compression. Compressors on the ordinary system, even if supplied with large air filters, would probably take in a considerable amount of dust from the air near the face, which would subject not only the compressor but also the drills to great wear.

With the Cummings system the make-up air only would be taken in from the outside, and the amount of this being small, the dust it contains could more effectually be caught in an air-filter. The make-up air could probably be supplied, in this case, by a small hand pump. It is to be noted, and this appeared also from some of the references given in the original paper, that rock drills have been successfully operated on the Cummings plan, and that tunnels have been driven economically by this method.

In the United States there seems recently to have been some appreciation of the utility of the return pipe system for operating drills in the manner described. A portable electrically driven dense air compressor for operating rock drills, manufactured in Colorado, U.S.A., is described and illustrated in the *Mining and Scientific Press* of the 9th of April, 1905.

It is possible that in this slightly altered form the scheme of Cummings for operating drills by his return system may yet come into extended use.

The application of the dense air system to the operation of drills in the manner described would deprive other dense air machinery, that might be installed, of the gain due to exhausting to drills operated in the ordinary way. The full gain in efficiency due to such exhausting would, however, rarely be obtainable in practice, as pointed out in the original paper, so that this argument would weigh little against the use of the system. The chief advantage of combination with the drills was that the drill pipe could be used as return for the dense air, so that there would be less obstruction of the shaft area. The economy due to the combination was only pointed out by the writer as an incidental advantage.

Mr. Schweder asks how the writer would vary the cutoff, and what he considered the shortest admissible cut-off, in a pair of single cylinder winding engines, without having the rope dance too much. In regard to the latter point, it must be remembered that the shortest cut-off, and therefore the least amount of periodic acceleration work takes place at the highest speed, when the regulating effect of the rotating and reciprocating masses join to produce an almost uniform torque at the periphery of the drum, differing widely from the torque diagram derived from the projected pressure ordinates of the indicator diagram. At starting, the effect will be the same as with an ordinary reciprocating engine.

In regard to the first part of the question, as to how the cut-off should be varied, the writer believes that this should be accomplished by some very static form of governor, so as to limit the speed below a fixed maximum. At the same time the maximum should be capable of being raised by making adjustments.

Mr. Schweder's plan of operating air-driven pumps in series seems feasible, and there might possibly be some slight gain in efficiency by arranging that the expanded and cooled air be re-heated by the heat of the mine. The efficiency of the mechanism, however, would be lower, particularly for the deepest pumps.

Speaking of the actual applications of the dense air system, of which the writer has seen one example, it should be understood, and this was also remarked upon by a well-known authority on compressed air, Mr. Frank Richards, that all the executed installations were equipped with the most primitive and insufficient kind of apparatus. Notwithstanding these defects, the efficiencies reached seem remarkable under the circumstances. With electric installations on the other hand, the greatest care has usually, and very properly so, been exercised in securing throughout the most efficient kind of apparatus.

In regard to pumping, the writer believes that electricty will, except in special cases, be the most universally employed means of operation, not so much on account of higher efficiency, but often because of convenience. This implies that other means of power transmission, like dense air, may sometimes be the most advantageous to adopt. No one system is the best for all cases. The Cummings dense air system, however, adds one more to the scant list of means for the transmission of power underground, and it would seem that a system which can stand the maltreatment and neglect, evident from the reports and correspondence on the Bisbee West plant, referred to in the original paper, and which can still show the efficiencies recorded, is well worth the careful attention and enquiry of all interested in the engineering problems connected with mine operation. Careful engineers, be they named electrical or mechanical, have fundamentally the same aims, and when called upon to decide what is best for existing conditions, will do so without prejudice and after careful examination of the merits of any device, purely mechanical or electrical, which seems available or suitable. They should bear in mind the broad definition of their class as given by the Institution of Civil Engineers: "An engineer is one whose profession it is to utilise the forces of Nature in the service of man."

Mr. W. Martin Epton: I beg to propose a vote of thanks to Mr. Behr. I am sure it afforded us all a great deal of instruction to listen to Mr. Behr, both in his original paper and in his reply this evening. It seems that the question of mine pumping comes down more or less to a question of transmission of power. As Mr. Behr remarks, one more means of transmission of power is brought before us by the Cummings system, and I am sure a great many of our Consulting Engineers in considering the matter of transmitting power down below will look into the Cummings system and be able to estimate very closely with the assistance of Mr. Behr's figures the comparative cost of installing it. I think this is one of the most valuable papers we have ever had read before this Institute.

Mr. Vaughan seconded. I think (he said) Mr. Behr has given a lesson to all of us, not only with this paper, but also with his previous great paper on Deep Level Winding, in the great care he has bestowed on his reply. It is very satisfactory to anyone who has contributed to the discussion on this paper to see the careful consideration that Mr. Behr bestows on any suggestions and criticisms that have been made. As an Institute we owe very great thanks to Mr. Behr, not only for his original paper, but for his exhaustive reply to the discussion that it provoked.

Mr. Behr: I thank you very much for the kind reception of my efforts, and I am very much obliged to you for the attention you have given me.

Robert Peele, 1908

Although not strictly in place in this chapter, reference may be made to what has been called the "two-pipe system" or "high-range compressed-air transmission," introduced some years ago by Charles Cummings.*

The machine or engine using the air makes in effect a closed circuit with the compressor. After the air has done its work in the motor cylinder, it is returned to the compressor at the pressure of the exhaust, through a second line of piping. The return pipe connects with a closed chamber at the compressor, in which the inlet valves are placed, thus enabling the compressor to begin its stroke with the cylinder filled under a considerable initial pressure. Then, after raising the pressure to the original point, the compressor delivers the air into the main, to be used again by the air engine. The actual working pressure of the air engine is, therefore, the difference between the pressures in the delivery and return pipes. Barring leakage, the same air is thus used over and over, the intention being that the compressor shall put back into the air kept in circulation the power expended in the motor engine cylinder.

Though the compressor itself is not materially different from the ordinary forms, the two-pipe system requires a rather complicated arrangement of piping and valves for charging the apparatus with air at the working pressure adopted, and for governing the speed and output according to the rate of consumption of air.† The advantages of the system are: a higher efficiency than is obtained from moderate-size compressors of the usual types, and less trouble from freezing at the motor engine by reason of the relative dryness of the air due to its higher tension. The efficiency increases with the pressure employed. In using compressed air without reheating the two-pipe system is superior in principle to the ordinary mode of operating compressed-air plant. But because of the greater first cost its advantages disappear when reheating can be adopted, and the single-pipe system is then found to be preferable.

The two-pipe system is best suited for machines working at full pressure throughout the stroke, such as machine drills or simple, direct-acting pumps. When the motor works expansively the pulsations become objectionable, as a regular flow of air is not maintained in the return pipe. Under these conditions the inertia and friction of high-pressure air in long pipe lines becomes noticeable and disadvantageous.

As the length of air pipe required for this system is doubled, not only may the first cost of the pipe go far toward offsetting the greater efficiency but, with at least twice as many joints in the pipe lines, the chances of loss from leakage are increased. And if very high pressures be used (pressures of several hundred pounds have been proposed), not only must the piping itself be heavier and more expensive, but the proportionate power loss from leakage is greater. For moderate distances, however, and when working at full pressure under the proper conditions, the foregoing disadvantages may be more than counterbalanced by the superior efficiency of the system. Though not yet in general use, the two-pipe system is said to have given satisfaction at several mines in New Mexico, Colorado, and California,* and has recently (1905) been proposed for use in the Johannesburg gold district. Some prominence is here given to the system because of its novel features and the probability that it may be found useful, if its disadvantages can be overcome. Reference may be made to a paper by H. C. Behr, published in 1905 in the *Transactions of the Mechanical Engineers' Association of the Witwatersrand*, in which the Cummings system is treated at length, with a discussion of its advantages as applied to compressed-air-driven pumps.

* Patent No. 456,941 was issued to Mr. Cummings in 1891.

† A detailed illustrated description is given by Frank Richards in *American Machinist*, April 28th, 1898, p. 23. See also *Compressed Air Magazine*, Oct., 1907, p. 4599.

* A. E. Chodzko, *Modern Machinery* (Chicago), Jan., 1899, p. 11.

Compressed Air Data, ed. William Lawrence Saunders
3rd Ed. c 1927, Compressed Air Magazine

THE RETURN-AIR SYSTEM

The principal difference between the displacement pump, Fig. 48, and the return-air system, Fig. 51, is that

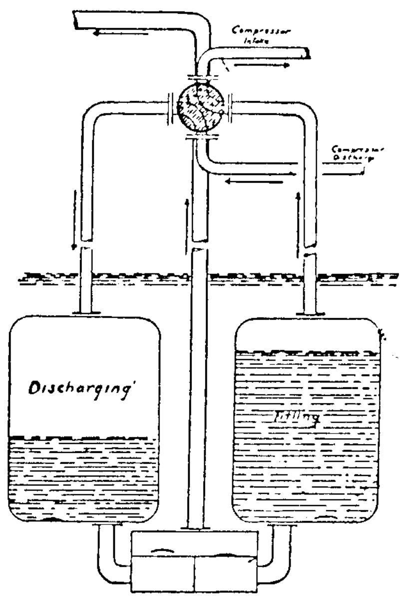

Fig. 51. Return-Air System.

the former method releases the air to atmosphere at practically full pressure after the pumping action, whereas the latter system returns this air to the compressor to be used over again. The return-air system, therefore, conserves most of this potential energy, whereas the displacement system throws it away. As a result, the return-air system will show an average efficiency of about 55%.

The essentials of the system are an air compressor driven by any convenient motive power; an automatic reversing switch in the compressor room; two air lines, each leading from the compressor through the switch to one pump tank; two tanks submerged in the fluid pumped, or within easy range of syphon action. Provision is, of course, made for automatically replacing the air which may be lost in the cycle by leakage, absorption, or in the operation of the switch. The single disadvantage of the system, as compared with the pneumatic displacement pump, is that a separate compressor must be used for the return-air pumping and it cannot be used for other purposes while pumping.

The principle of the return-air system is very simple. Compressed air is admitted to a tank full of fluid, forcing the fluid out through a suitable discharge pipe, its return being prevented by a check valve. The air which has displaced the fluid from the tank is then drawn back through the air line and switch, through the compressor intake valves, the compressor cylinder and the discharge valves, until equilibrium is secured throughout the system which then contains a charge of air at a certain pressure above atmosphere. This equalizing operation takes but a short time during which the compressor operates at no load, pressures being balanced on both sides of its piston. The moment that equilibrium is attained the compressor takes up the load, compressing the air in the second tank and drawing its intake, already at high pressure, from the first tank and pipe line. As pressure increases in the second tank, the fluid is discharged; while as pressure diminishes in the first tank, the fluid enters.

The Engineering Magazine
January 1914

One characteristic of the ordinary plain displacement pump is the waste of power entailed by the direct release of the displacing air volume after the fluid is rejected from the pump tank. This air, after doing its work, is still at practically full pressure, therefore having all its potential energy of expansion. Its direct exhaust into the atmosphere after displacement is the throwing away of this expansive power without any useful effect.

If the air pipes from the compressor to the tanks are carried back to the compressor room to apply the pressure remaining in the air volume after displacement on the reverse side of the compressor piston, where it would help in compressing the air into the pump tank, we have a "return-air" expansive displacement pump, operating at good economy.

This system has all the advantages of simplicity that the non-expansive system enjoys, with the additional advantage of being much more efficient. Under normal conditions it is safe to say that the "return-air" system will show an average efficiency of 50 to 55 per cent, this efficiency being the ratio of horse power of water lifted to indicated horse power in the steam cylinder of the compressor, including all losses.

POWER

February 16, 1915

Return-Pipe Compressed-Air Practice

By Frank Richards

A letter from a California correspondent asks why it is that more has not been made of the Cummings system of compressed-air power transmission. He says that from the results which have been actually attained by the system it could be advantageously employed in many places, especially as, besides the economy of it, there is no danger of fire or explosion, and it can be operated under water.

Notwithstanding that the return-air or two-pipe pumping system, for raising water by the direct pressure of air, is quite extensively and successfully employed in different parts of the country, and that this system has been fully described in various publications, the essential principles of the Cummings system in its entirety are not generally well understood even where it happens to be known at all. Patented a full generation ago, it seems to have been exploited mostly in California, and it may be worth while to call the attention of power users to it again.

It is rather curious that the new departure which this system represents—the use of higher pressures—is quite in line with the improvements in steam engines, in oil engines, especially of the Diesel type, and in electrical practice. It may be claimed, however, that the two-pipe

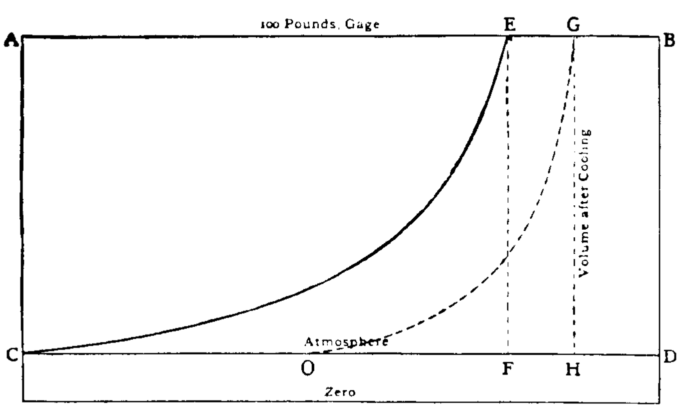

Fig. 1. Air between 0 and 100 Lb. Gage

air system "goes them one better." In the compound- or the triple-expansion steam engine it seems to be the last added portion of the pressure which secures the economy, but the entire range of the pressure from the bottom to the top has all been retained, while the compressed-air system here to be spoken of retains and uses only the higher, and presumably more profitable, range of pressure.

The essential feature of the system is the constant maintenance of a high pressure upon the air employed. Instead of continually compressing fresh atmospheric air up to, say 100 lb. gage, using it in the motor at that pressure, with or without expansion, and then exhausting the air into the atmosphere again, a constant intake pressure of, say 100 lb. is maintained at the compressor. The air is compressed to, say 200 lb., is transmitted to and is used in the motor at that pressure, and then is exhausted and carried back to the compressor at a pressure of 100 lb., to be compressed and used again, and so on.

Different Pressure Ranges Compared

The accompanying diagrams, Figs. 1, 2, 3, are all drawn to the same scale for equitable comparison, and may be studied together, although each represents an operation entirely distinct from and unrelated to the others; that is, they are not successive stages of one operation. In each case the same volume of air fills the cylinder at the beginning of the compression, but the actual weights

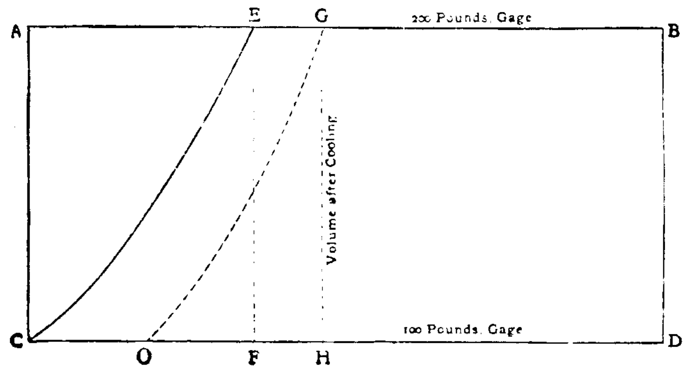

Fig. 2. Air between 100 and 200 Lb. Gage

or quantities of air are very different, only Fig. 1 beginning the compression with "free air," or air at atmospheric pressure.

Fig. 1 represents the adiabatic compression of a given volume of air from atmospheric pressure, say 15 lb. to the inch absolute, to a gage pressure of 100 lb., or 115 lb. absolute. Fig. 2 shows the compression of an equal volume (not an equal weight) of air, but under an initial pressure of 100 lb. gage, to a delivery pressure of 200 lb.; and in Fig. 3 an equal volume of air at 200 lb. is compressed to 300 lb.

In each case the initial volume of air compressed is represented by the area of the rectangle *ABDCA*. When the air has been compressed to the gage pressure specified in each case its volume is represented by the area *EBDFE*, and this will be the volume assumed to be discharged into the pipes and receiver. As we are speaking now from the purely theoretical viewpoint, nothing is said about clearance or other allowances made in practice.

It is well understood that the operation of compression invariably increases the temperature of the air very much, but this temperature it is impossible to maintain, and unless reheating is employed, the air is never used at the high temperature at which it is delivered by the compressor. As the air cools to normal temperature before it is used, its volume being reduced proportionately, the actual volume available for use is represented by the area *GBDHG*, this being in Fig. 1 only about an eighth of the initial volume, and not much more than one-half the volume *EBDFE*, as delivered by the compressor.

The air delivered under either compression represented may be said to have equal working value, volume for volume, the available pressure being 100 lb. in either case, the air in Fig. 1 at 100 lb. working against atmosphere only, the air in Fig. 2 at 200 lb. working against

a back pressure in the return pipe of 100 lb., and that in Fig. 3 at 300 having a back pressure of 200 lb.

In compressing air from 100 to 200 lb., as in Fig. 2, the temperature of the air is not raised nearly as much as in Fig. 1 and, consequently, the shrinkage in cooling from volume *EBDFE* to volume *GBDHG* is proportionately much less than in Fig. 1. The volume *GBDHG* here

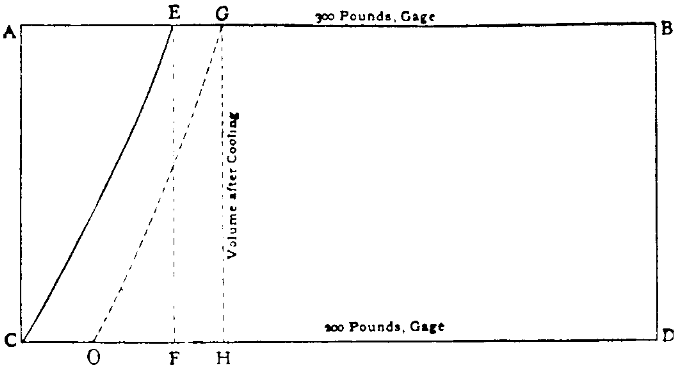

FIG. 3. AIR BETWEEN 200 AND 300 LB. GAGE

available for work is more than one-half the initial volume *ABDCA*, or four times the volume available in Fig. 1. At the same time it is to be noted that the mean effective pressure in the compressor cylinder for the stroke, which is the measure of the actual work of compression, is decidedly less than double that of Fig. 1. Getting fully four times the available volume for less than double the power employed certainly looks like doubling the efficiency by halving the relative cost of the compression.

In Fig. 3, compressing the air from 200 to 300 lb., the heating of the air is still less and the consequent shrinkage by cooling also is less. The available volume delivered, *GBDFG*, is five times the corresponding volume in Fig. 1, while the mean effective pressure required for the compression and delivery of the air is less than 2.1 times as much, which seems to be decidedly more than doubling the efficiency.

It has been assumed in each case above that the initial air temperature is 60 deg. F. With the same increase of 100 lb. in pressure the final temperatures will be 485, 163 and 121 deg., the rise of temperature being, respectively, 425, 103 and 61 deg. The enormous rise of temperature in compressing from atmospheric pressure has led to the general adoption of two-stage compression, with intercooling of the air, thereby gaining something in economy, avoiding the overheating of the surfaces, the burning of the lubricants and the danger of fires and explosions. With the heating that occurs in Figs. 2 and 3 there is no necessity for employing the two-stage compressor, and little possibility of any increased economy through its employment.

The ratio of final and initial absolute pressures is: In Fig. 1, 7.666; in Fig. 2, 1.869; in Fig. 3, 1.465. The ratio of the volume after cooling to 60 deg., or the volume available for use, to the initial volume is: In Fig. 1, 0.1304; in Fig. 2, 0.535 and in Fig. 3, 0.6825. The relative costs of compression, as measured by the power used, or the mean effective pressures for the compression divided by the volume after cooling, are: In Fig. 1, $11.6 \div 0.1304 = 319$; in Fig. 2, $78.88 \div 0.535 = 147$; and in Fig. 3, $86.83 \div 0.6825 = 127$. Here the ratio of the cost in Fig. 1 is $319 \div 147 = 2.17$, and of Fig. 1 to Fig. 3 it is $319 \div 127 = 2.51$.

It is understood that wherever this air is used—that is, the air of Fig. 2 and Fig. 3—whether for driving a rock drill, for a steam pump or an air motor of any kind, the air instead of being discharged into the atmosphere, as it would be from Fig. 1, is piped back to the compressor with only 100 lb. of its pressure used; then, volume for volume, the air used would be of the same power value in either case, if not used expansively. As the available volume delivered as shown in Fig. 2 is four times that in Fig. 1, a compressor of one-fourth the capacity, or, at equal piston speeds, with a cylinder one-half the diameter, will be sufficient for the work. The maximum unbalanced pressure against the piston would be no greater in one case than in the other, only it would be continued for a longer or a shorter portion of the stroke. There would be no additional strength required in any of the working parts of the machine, except that the air cylinder and connections would have to be strong enough for the maximum pressure.

As the same air is used over and over again in the two-pipe system, arrangements being provided for making up leakage losses, there is no appreciable accumulation of moisture and no possibility of freezing up, even if sufficiently low temperatures should occur, which they do not. At the same time more or less of the lubricant is carried back and forth in the air and comes in contact with the working surfaces. As the system is a closed one, being entirely out of touch with the surrounding atmosphere and not affected by the local pressure, it will work at one altitude just as well as at another.

WHY THE SYSTEM'S USE HAS BEEN LIMITED

Now as to why the system has not been more extensively employed; there is the fact to begin with that even yet it is not generally as well known and understood as it should be. Then, evidently, it would not be likely to be much used for intermittent work, such as the driving of rock drills which are continually changing their location, and where the maintenance of the return connection would cost in time and trouble enough to cancel the prospective advantage.

Apparently, the best employment of the system would be for the driving of ordinary steam pumps where constant pressure is usually required for practically the entire stroke. The air of Fig. 2, at 200 lb. pressure and 100 lb. back pressure, or the air at the higher pressures of Fig. 3 does not permit much profitable expansion in use. When used for rotative purposes in an engine or motor, the cutoff, as the compression diagram suggests, should never occur earlier than three-quarter stroke, so that the cutoff that may be accomplished by a good slide-valve engine would be all that would be available in any case. In this respect the air in Fig. 1 would have some advantage, as, to secure the greatest economy, it should be cut off before half-stroke, and a certain saving would be accomplished by the expansion which would not be possible where the higher pressures were employed.

There is a necessity for the compressor supplying the air and the engine or motor using the air to approximately keep pace with each other, not necessarily stroke for stroke, but so that, with the aid of suitable receiver capacity, the delivery and the return air pressures shall be maintained as constant as possible. This implies that the two working units of the system should be adapted to each other in capacity and that an automatic pressure governor should control the compressor.

TABLE 10.30. THEORETICAL HORSEPOWER REQUIRED AT ALTITUDE TO COMPRESS 100 CU. FT. OF FREE AIR PER MIN.

Altitude, ft	Isothermal compression					Adiabatic compression								
	Single- and two-stage					Single-stage			Two-stage					
	Gauge pressure					Gauge pressure			Gauge pressure					
	60	80	100	125	150	60	80	100	60	80	100	125	150
0	10.4	11.9	13.2	14.4	15.5	13.4	15.9	18.1	11.8	13.7	15.4	17.1	18.7
1,000	10.2	11.7	12.9	14.1	15.1	13.2	15.6	17.8	11.6	13.5	15.1	16.8	18.3
2,000	10.0	11.4	12.6	13.8	14.8	13.0	15.4	17.5	11.4	13.2	14.8	16.4	17.9
3,000	9.8	11.2	12.3	13.5	14.4	12.8	15.2	17.2	11.2	13.0	14.5	16.1	17.5
4,000	9.6	11.0	12.1	13.2	14.1	12.6	14.9	16.9	11.0	12.7	14.2	15.7	17.1
5,000	9.4	10.7	11.8	12.8	13.7	12.4	14.7	16.5	10.8	12.5	13.9	15.4	16.7
6,000	9.2	10.5	11.5	12.5	13.4	12.2	14.4	16.2	10.6	12.2	13.6	15.1	16.4
7,000	9.0	10.3	11.2	12.2	13.0	12.0	14.2	16.0	10.4	12.0	13.4	14.8	16.0
8,000	8.9	10.0	11.0	11.9	12.7	11.8	14.0	15.7	10.2	11.8	13.1	14.5	15.6
9,000	8.7	9.8	10.7	11.6	12.4	11.6	13.7	15.4	10.0	11.6	12.8	14.1	15.3
10,000	8.5	9.6	10.4	11.4	12.1	11.5	13.5	15.1	9.8	11.3	12.6	13.8	15.0

TABLE 10.31. APPROXIMATE BRAKE HORSEPOWER REQUIRED BY AIR COMPRESSORS

Figures given are bhp per 100 cu. ft. of free air per min. actually delivered

Altitude, ft.	Single-stage			Two-stage			
	Psig			Psig			
	60	80	100	60	80	100	125
0	16.3	19.5	22.1	14.7	17.1	19.1	21.3
1,000	16.1	19.2	21.7	14.5	16.8	18.7	20.9
2,000	15.9	18.9	21.3	14.3	16.5	18.4	20.5
3,000	15.7	18.6	20.9	14.0	16.1	18.0	20.0
4,000	15.4	18.2	20.6	13.8	15.8	17.7	19.6
5,000	15.2	17.9	20.3	13.5	15.5	17.3	19.2
6,000	15.0	17.6	20.0	13.3	15.2	17.0	18.8
7,000	14.7	17.3	19.6	13.0	14.9	16.6	18.4
8,000	14.5	17.1	19.3	12.7	14.6	16.2	18.0
9,000	14.3	16.8	18.9	12.5	14.3	15.9	17.6
10,000	14.1	16.5	18.6	12.3	14.1	15.6	17.2
12,000	13.6	15.9	17.9	11.8	13.5	15.0	16.5
14,000	13.1	15.2	17.2	11.3	12.9	14.3	15.7

Note: Bhp per 100 cu. ft. of free air per min. will vary considerably with the size and type of compressor.

Edited by
John P. Rollins
Professor of Mechanical Engineering
Clarkson College of Technology

TABLE 10.32. THEORETICAL HORSEPOWER REQUIRED TO COMPRESS AIR FROM ATMOSPHERIC PRESSURE TO VARIOUS PRESSURES—MEAN EFFECTIVE PRESSURES (mep)

Discharge pressure			Isothermal compression, single or multistage		Adiabatic compression*				Theoretical intercooler gauge pressure	Per cent of power saved by two-stage over single-stage adiabatic compression
					Single-stage		Two-stage			
Psig	Psia	Atm abs	Mep	Theoretical hp per 100 cu. ft.	Mep	Theoretical hp per 100 cu. ft.	Mep, psi referred to low-press. air cylinder	Theoretical hp per 100 cu. ft.		
5	19.7	1.34	4.13	1.8	4.48	1.96				
10	24.7	1.68	7.57	3.3	8.21	3.58				
15	29.7	2.02	10.31	4.5	11.4	5.0				
20	34.7	2.36	12.62	5.5	14.3	6.2				
25	39.7	2.70	14.68	6.4	16.9	7.4				
30	44.7	3.04	16.30	7.1	19.2	8.4				
35	49.7	3.38	17.90	7.8	21.4	9.3				
40	54.7	3.72	19.28	8.4	23.4	10.2				
45	59.7	4.06	20.65	9.0	25.2	11.0				
50	64.7	4.40	21.80	9.5	27.0	11.8				
55	69.7	4.74	22.95	10.0	28.7	12.6				
60	74.7	5.08	23.90	10.4	30.3	13.3				
65	79.7	5.42	24.80	10.8	31.9	13.9				
70	84.7	5.76	25.70	11.2	33.3	14.6	29.2	12.8	20.6	12.3
75	89.7	6.10	26.62	11.6	34.7	15.2	30.3	13.3	21.6	12.5
80	94.7	6.44	27.52	12.0	36.0	15.7	31.3	13.7	22.7	12.7
85	99.7	6.78	28.21	12.3	37.3	16.3	32.3	14.1	23.6	13.5
90	104.7	7.12	28.93	12.6	38.6	16.9	33.2	14.5	24.5	14.2
95	109.7	7.46	29.60	12.9	39.8	17.4	34.2	14.9	25.5	14.4
100	114.7	7.80	30.30	13.2	40.9	17.9	35.0	15.3	26.3	14.5
110	124.7	8.48	31.42	13.7	43.2	18.9	36.7	16.1	28.1	14.8
120	134.7	9.16	32.60	14.2	45.2	19.8	38.3	16.8	29.8	15.1
130	144.7	9.84	33.75	14.7	47.2	20.7	39.6	17.3	31.5	16.4
140	154.7	10.52	34.67	15.1	49.2	21.5	40.8	17.9	32.9	15.7
150	164.7	11.20	35.59	15.5	51.0	22.3	42.3	18.5	34.5	17.1
160	174.7	11.88	36.30	15.8	....	....	43.6	19.0	36.1	
170	184.7	12.56	37.20	16.2	....	....	44.7	19.5	37.3	
180	194.7	13.24	38.10	16.6	....	....	45.8	20.0	38.8	
190	204.7	13.92	38.80	16.9	....	....	46.8	20.4	40.1	
200	214.7	14.60	39.50	17.2	....	....	47.8	20.9	41.4	
250	264.7	18.00	42.70	18.6	....	....	52.5	22.7	47.6	
300	314.7	21.40	45.30	19.7	....	....	56.5	24.5	53.4	
350	364.7	24.81	47.30	20.6	....	....	59.6	26.1	58.5	
400	414.7	28.21	49.20	21.4	....	....	62.7	27.4	63.3	
450	464.7	31.61	51.20	22.3	....	....	65.3	28.6	67.8	
500	514.7	35.01	52.70	22.9	....	....	67.8	29.6	71.2	
550	564.7	38.41	53.75	23.4	....	....	70.0	30.6	76.3	
600	614.7	41.81	54.85	23.9	....	....	72.3	31.3	80.5	

*Based on a value for n of 1.3947.

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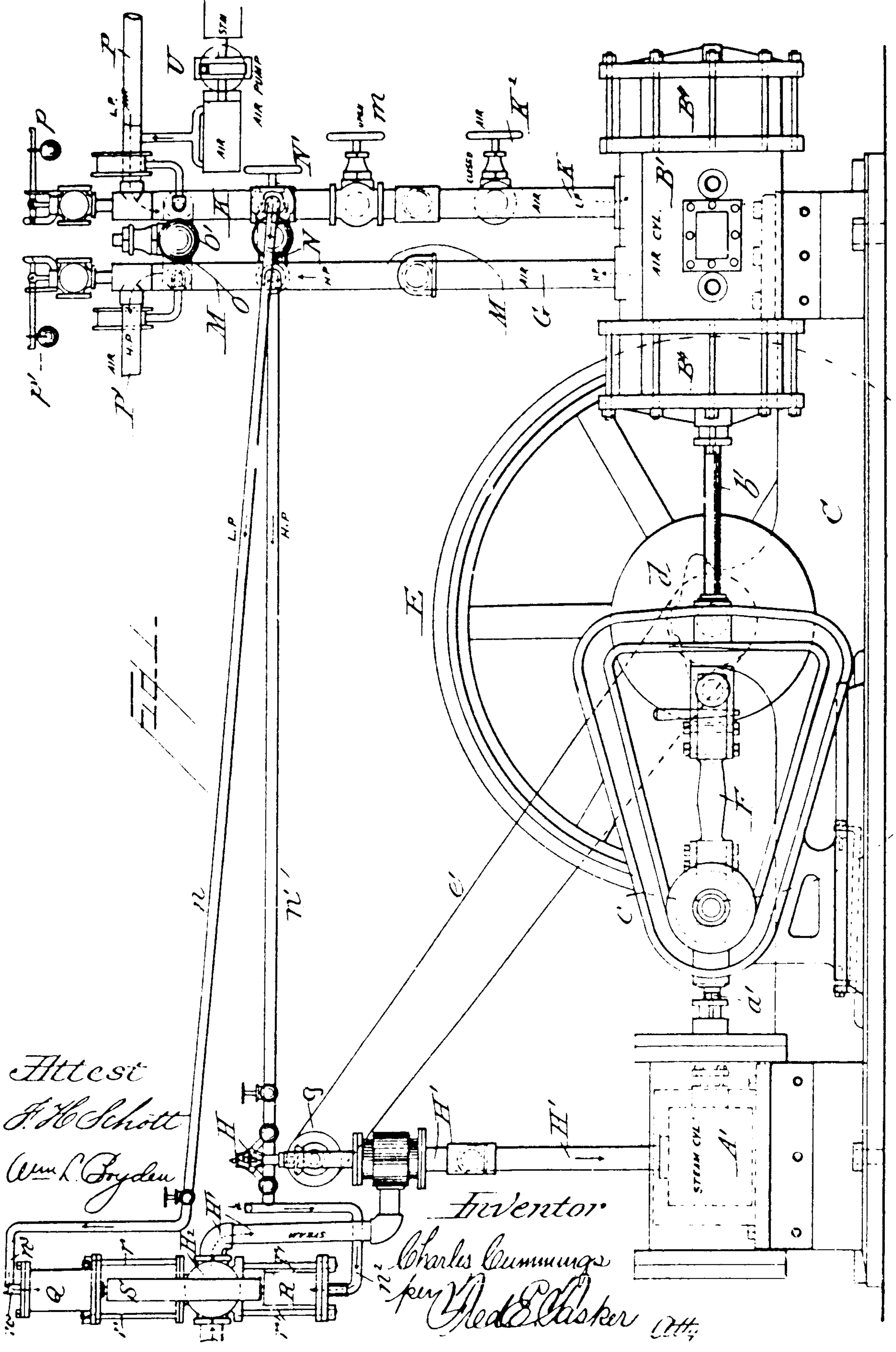
C. CUMMINGS.

APPARATUS FOR TRANSMITTING POWER BY MEANS OF COMPRESSED AIR.

No. 456,941. Patented Aug. 4, 1891.



Attest
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Wm. L. Boyden

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Charles Cummings
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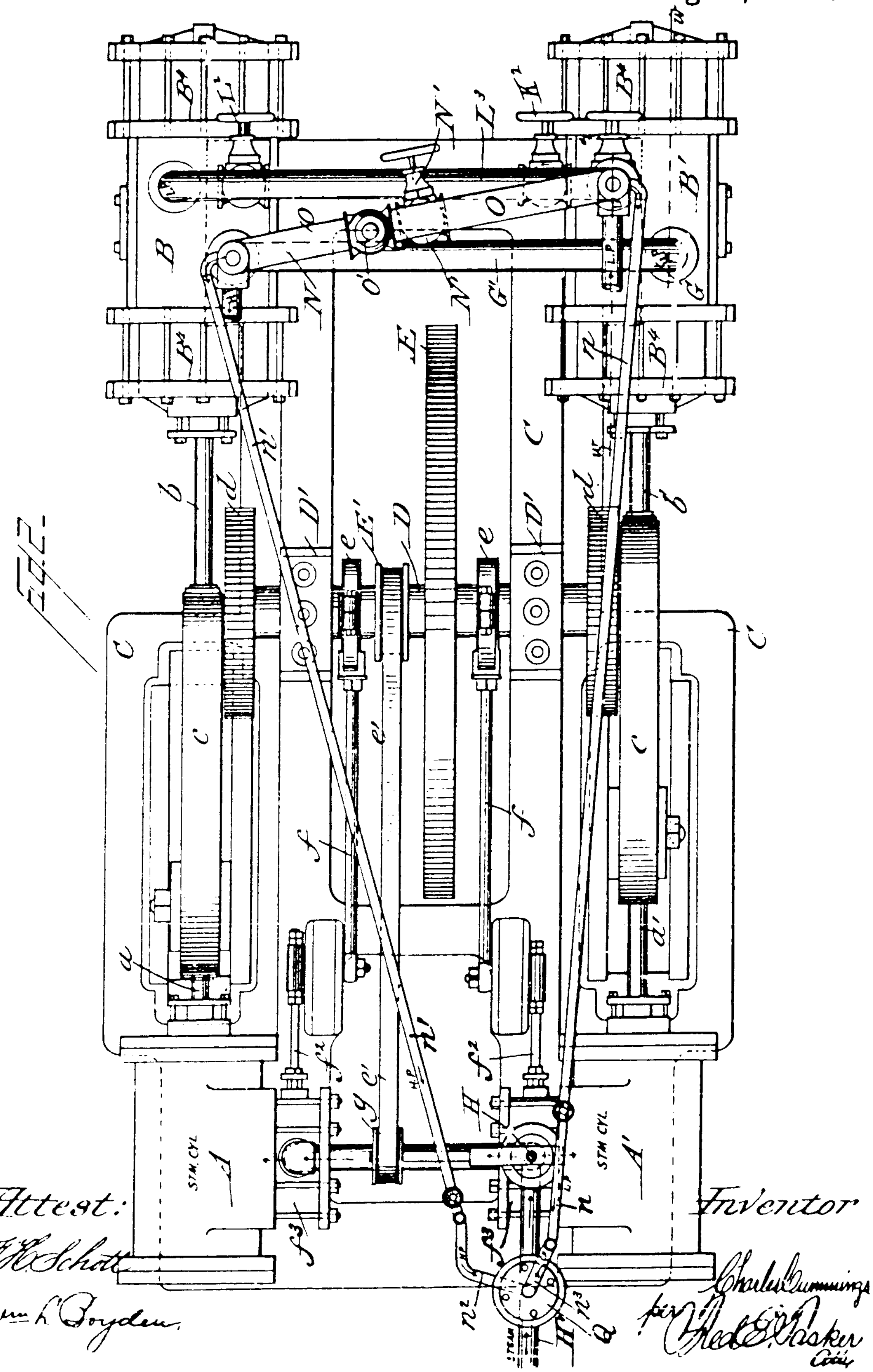

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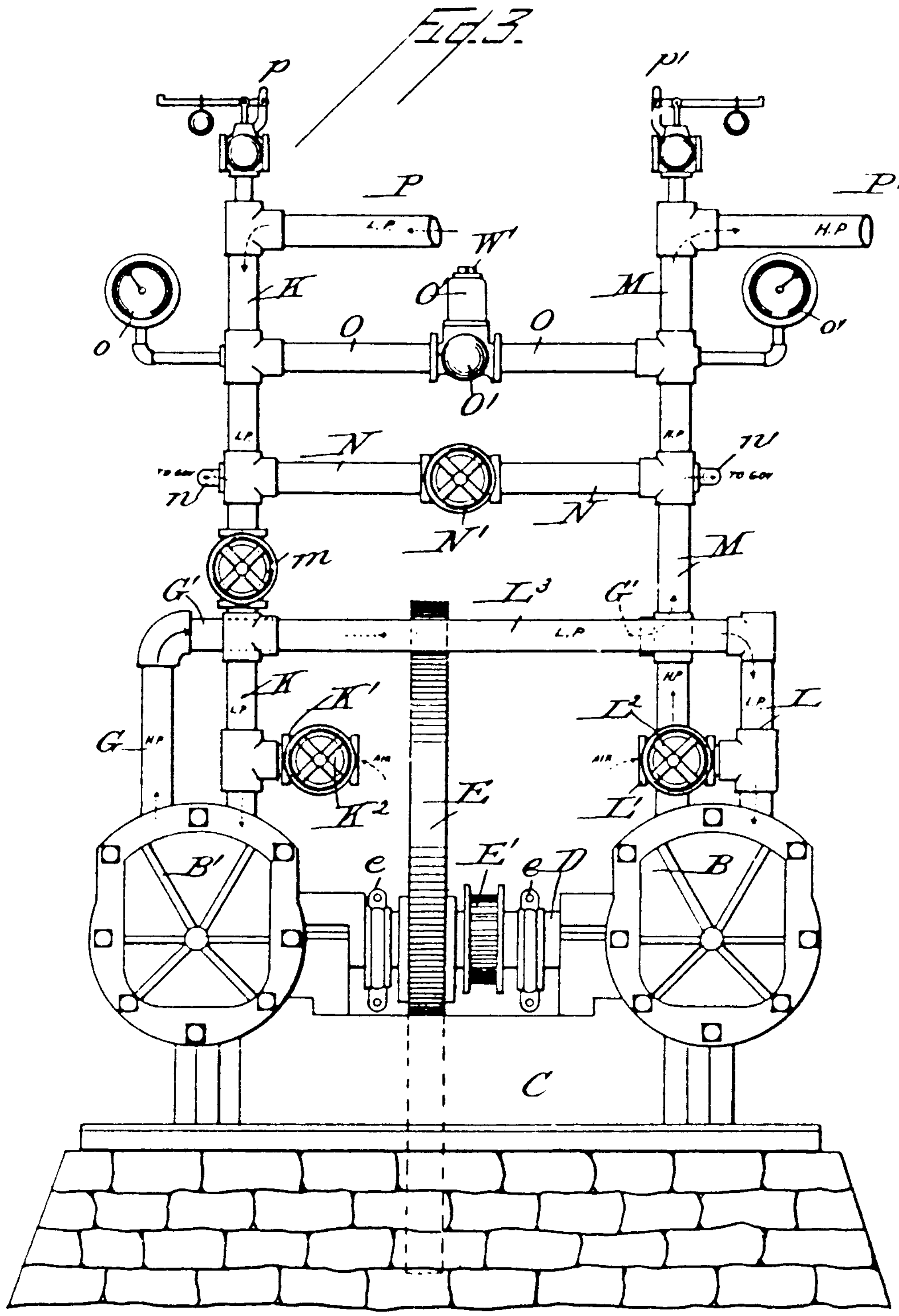

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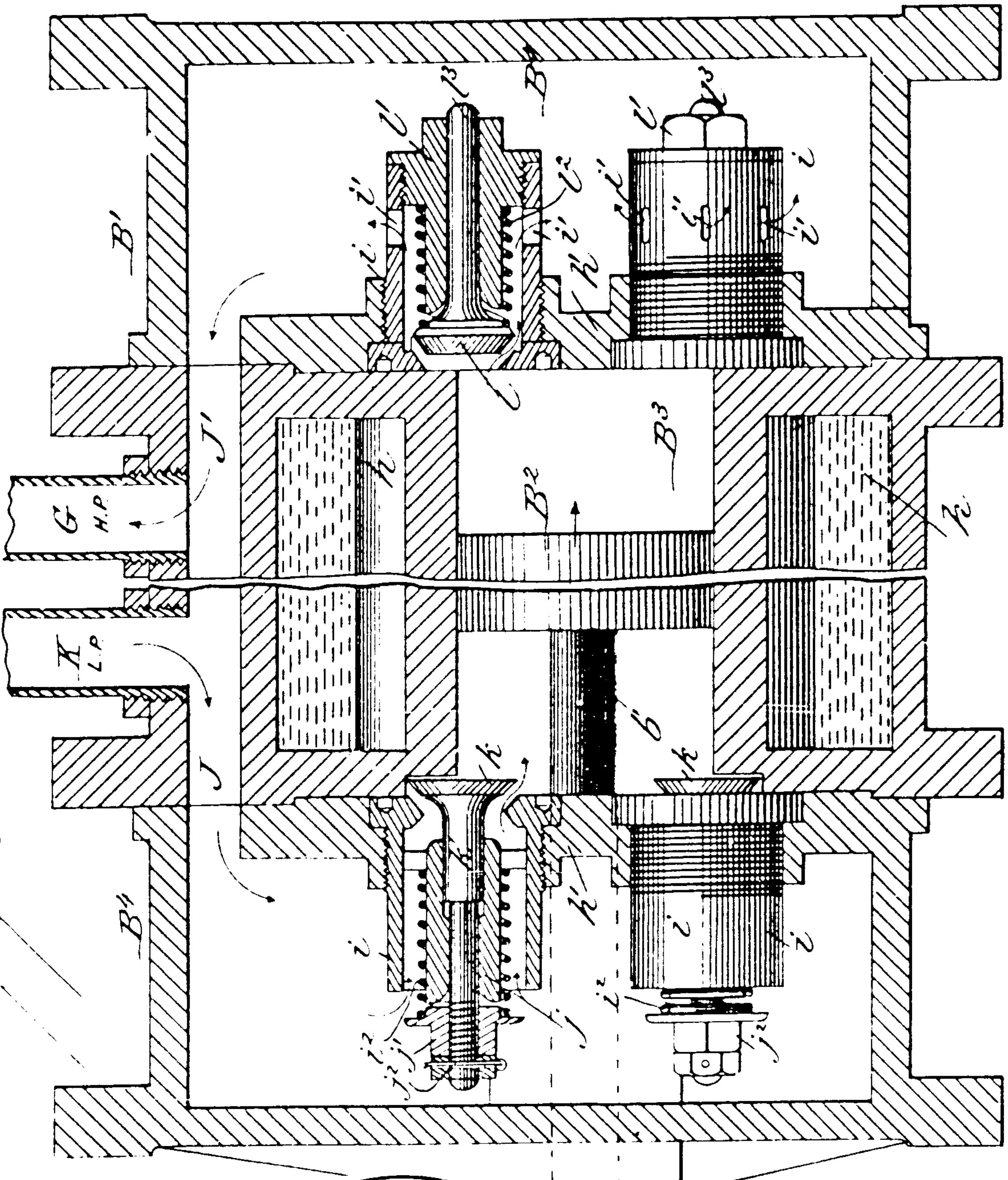

Fig. 4.

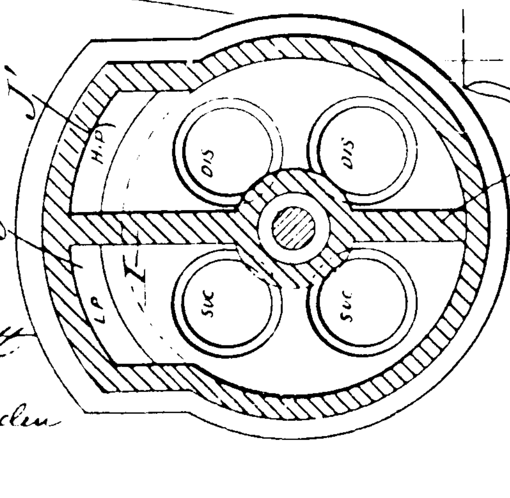

Fig. 5.

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C. CUMMINGS.

APPARATUS FOR TRANSMITTING POWER BY MEANS OF COMPRESSED AIR.

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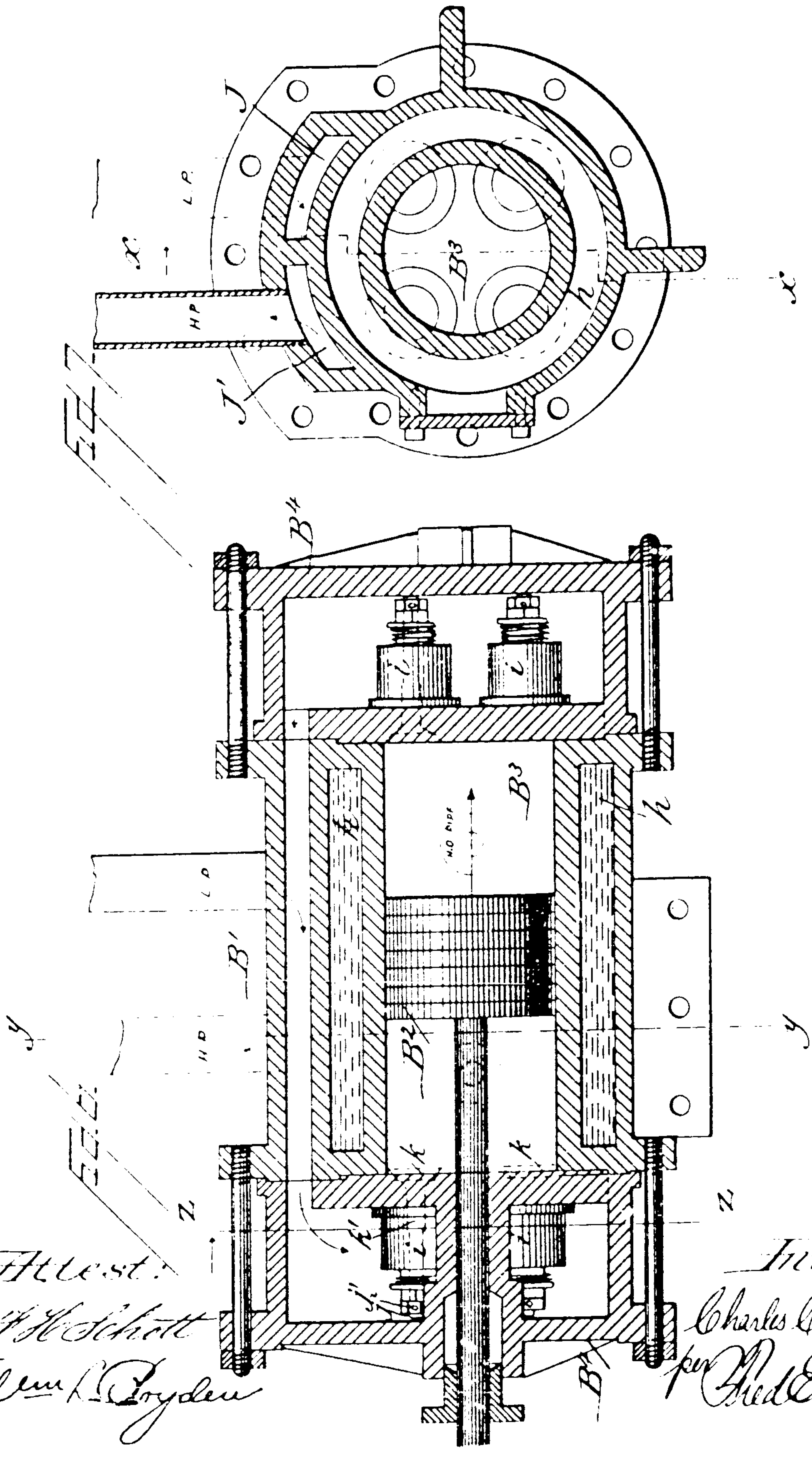

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C. CUMMINGS.

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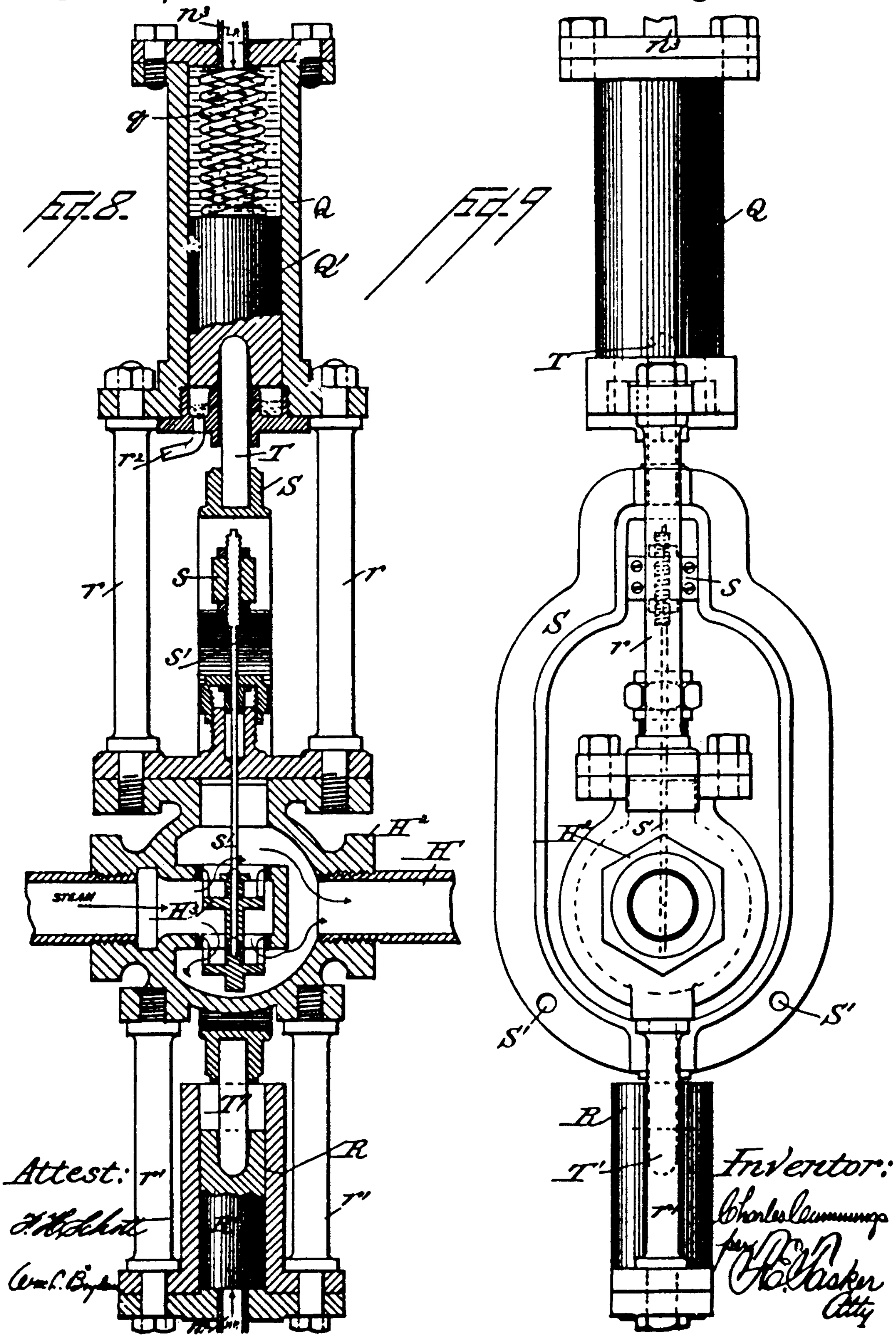

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APPARATUS FOR TRANSMITTING POWER BY MEANS OF COMPRESSED AIR.

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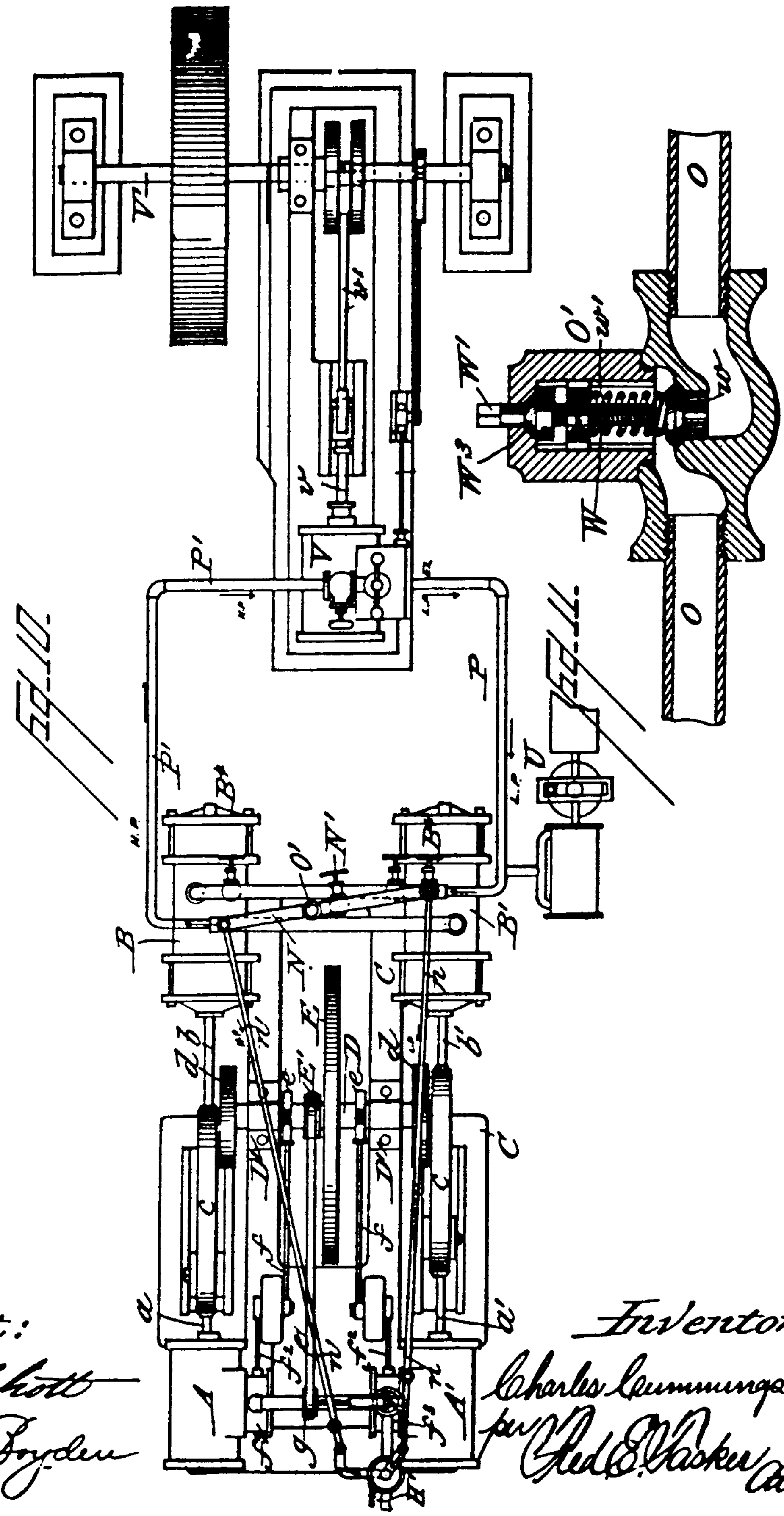


Attest:

F. H. Schott

Wm. L. Boyden

Inventor:

Charles Cummings

per Fred E. Tasker Atty

UNITED STATES PATENT OFFICE.

CHARLES CUMMINGS, OF OAKLAND, CALIFORNIA.

APPARATUS FOR TRANSMITTING POWER BY MEANS OF COMPRESSED AIR.

SPECIFICATION forming part of Letters Patent No. 456,941, dated August 4, 1891.

Application filed June 12, 1890. Serial No. 355,228. (No model.)

To all whom it may concern:

Be it known that I, CHARLES CUMMINGS, a citizen of the United States, residing at Oakland, in the county of Alameda and State of California, have invented certain new and useful Improvements in Apparatus for Transmitting Power by Means of Compressed Air; and I do hereby declare the following to be a full, clear, and exact description of the invention, such as will enable others skilled in the art to which it appertains to make and use the same.

This invention relates to machinery for transmitting power by means of compressed air or other gases, the object of the invention being to provide better, cheaper, more compact, and more efficient mechanism for the purpose indicated, whereby the best results may be achieved in the application of energy to successful work at a point distant from where the energy is generated with the least possible percentage of loss of energy in the transmission; and the invention consists, essentially, in the construction, arrangement, and combination of parts for carrying into practical effect and operation the underlying principle or method upon which the invention is based, which is—to wit, the employment of two unequal pressures of air, both above the normal atmospheric pressure and maintained at a constant ratio to each other, the air circulating through a system of pipes or conduits and other parts, which system is closed to the external atmosphere after the apparatus has once been charged, and said confined body of air being alternately expanded and compressed during the circulation, all substantially as will be hereinafter fully described, and then more particularly pointed out in the subjoined claims.

In the annexed drawings, illustrating my invention, Figure 1 is a side elevation of my improved machine for transmitting power by means of compressed air or other gases. Fig. 2 is a top plan view of the same. Fig. 3 is a right-hand end elevation. Fig. 4 is an enlarged longitudinal section on the line *w w* of Fig. 2. Fig. 5 is a cross-section on the line *z z* of Fig. 6. Fig. 6 is a longitudinal section on the line *x x* of Fig. 7. Fig. 7 is a cross-section on the line *y y* of Fig. 6. Fig. 8 is a vertical section of the air governor or device for maintaining the unequal air-pressures at a fixed and constant ratio. Fig. 9 is a side elevation of the same. Fig. 10 is a plan view of the entire apparatus on a diminished scale and shows an example of driven machinery. Fig. 11 is a vertical section of the automatic valve which is sometimes used.

Like letters of reference designate corresponding parts throughout all the different figures of the drawings.

My invention aims by novel means to practically embody a novel principle or method which has never to my knowledge been carried into actual operation in a practical machine. This principle or method consists in the use of two unequal pressures of air, each above the normal atmospheric pressure and bearing to each other a fixed ratio, which is kept constant, the body of air which is held at these two pressures being confined in a closed system. It will be remembered that in the common and well-known machine for transmitting power by means of compressed air the air is drawn into the compressing-engine from the open atmosphere, after which it is compressed until its expansive force is sufficient to perform the intended duty where the power is required. The compressed air, after acting on the motor or engine where the power is applied, is exhausted back into the atmosphere, thus falling again to the normal atmospheric pressure.

In my improved apparatus I employ two air-pressures, both above the atmospheric pressure instead of one increased pressure and the ordinary atmospheric pressure, as in the common machine, and in my apparatus the air used is not drawn from the atmosphere or exhausted thereinto during the operations of the apparatus after it has once been charged, except for the incidental purpose of supplying the slight loss occasioned through leakage; but the operations of the apparatus are performed on a confined body of air shut off from communication with the external atmosphere. I have a suitable driving apparatus or power device which actuates the air-compressor. This compressor is so constructed and works in such a way as to create two pressures of air, both above the ordinary atmospheric pressure, one being, for example, one hundred pounds to the square

inch and the other, for example, two hundred pounds to the square inch, the high pressure in this case being double the low pressure, or the ratio as two to one. There is a system of pipes provided with valves, which pipes carry the air to the driven apparatus, where the compressed air is to do its work. The air circulates constantly between the compressor and the motor, passing through reservoirs or receivers, if desired, and in this driven machine the unequal air-pressures will be on opposite sides of the piston, thus leaving the piston unbalanced and permitting it to move in the ordinary way. Hence in one part of this circulating system the air is at one pressure, while in another part it is at another pressure, the air thus being subject to constant change from one pressure to another as it circulates. This change takes place alternately, the air, when at high pressure, doing its work in the motor and then exhausting into the low-pressure air and so continuing until it again reaches the compressor and is reconverted into high-pressure air. This change of volume takes place with but little loss of power through heat, so that efficiency in the operation of the machine is promoted.

C denotes the main frame or bed of the machine, which may be of any suitable and desirable form to permit the convenient arrangement thereon and therewith of the several mechanical parts.

B B′ designate the two parallel cylinders of the air-compressor. I preferably use two, although there may be any number. The air-compressor is driven by any suitable means. Steam is the motive power in the example of mechanism illustrated in the drawings.

A and A′ denote the steam-cylinders, which are of any usual pattern. The piston of cylinder A has piston-rod *a*. Cylinder A is, for the sake of convenience, located in line with the air-cylinder B of the air-compressor. This latter cylinder has its piston provided with piston-rod *b*, which is in line with the piston-rod *a*. Between the piston-rods *a* and *b* is the yoke *c*, which is firmly connected to both rods. The piston of cylinder A′ has piston-rod *a′*. Cylinder A′ is, for the sake of convenience, located in line with the air-cylinder B′ of the air-compressor. This latter cylinder has its piston provided with piston-rod *b′*, which is in line with the piston-rod *a′*. Between the piston-rods *a′* and *b′* is the other yoke *c*, which is firmly connected to both rods.

The main shaft D of the steam-engine is carried horizontally in suitable bearings D′ D′ on the main frame C. At each end of this main shaft are the crank-disks *d d*, having their crank-pins set at right-angles, so that the engine may start when the connecting-rods are in any position, to which are pivoted the connecting-rods F F, which are connected at their other ends at points within the yokes *c c*. (See Fig. 1.) On the drive-shaft D are also the band-wheel E, governor-pulley E′, and eccentrics *e e*, said eccentrics having the eccentric-rods *f f*, that are connected to the valve-rods f^2 f^2, belonging to the valves within the valve-chests f^3 f^3 on the sides of the two steam-cylinders A and A′. The belt *e′*, passing around the governor-pulley E′, runs also over the pulley *g* on the shaft of the ball-governor H, (see Figs. 1 and 2,) which is of any common and well-known construction for a steam-governor and which serves the usual purpose by controlling a valve in the steam-pipe H′, which runs from the boiler to the steam-cylinders, passing on its way through the air-governor, as shown in Fig. 1, the construction of which air-governor will be presently explained.

The ball-governor is used to avoid accidents in case anything happens to the air-governor, and is of course adjusted properly for this purpose. I have thus given a summary description of the steam-engine, which in the present example of my invention I have arranged for driving the air-compressor. It will be understood, and I here emphatically state, that said compressor may be operated by any motor, if desired, as well as by a steam-engine. Therefore the engine herein shown and described is simply given as one convenient type or form of mechanism for the purpose.

I will now proceed to describe the construction of the mechanism for compressing the air in such a manner as to produce the two unequal air-pressures, both above the normal atmospheric pressure, which I have already above alluded to as being the foundation idea of the invention.

B and B′ represent, as we have seen, the two air-cylinders, within which are the pistons B^2, one of which has the piston-rod *b* and the other the piston-rod *b′*. By referring to Figs. 4, 5, 6, and 7 the construction of these air-cylinders will be clearly seen. They are constructed similarly.

B^3 denotes the bore of the cylinder, within which the compressing-piston B^2 is fitted and wherein it operates. This bore is preferably surrounded with a water-jacket *h* for the purpose of keeping the parts from becoming heated during the operation of compressing the air. The cylinders are provided at each end with hollow heads B^4 B^4, which are bolted or otherwise firmly secured to the main body of the cylinder, as shown in Fig. 6. Each of the hollow heads B^4 is divided by a vertical partition I (see Fig. 5) into two compartments, and in the main body of the cylinder, above the water-jacket *h*, are two horizontal channels J and J′, divided by a longitudinal partition or wall exactly in line with the partitions I I in each head, so that the chambers in the heads, in conjunction with these channels J and J′, which communicate therewith, form two separate and distinct longitudinal compartments. The bore B^3 has the heads *h′* *h′*, which constitute the inner vertical walls of the hollow heads B^4, being an integral

part thereof, each of which heads or walls h' is provided with four openings, (see Fig. 5,) two of which lie on one side of the partition I and two on the other side. At each of these openings a valve is arranged. The hollow heads are of sufficient size to contain the mechanism of these valves, as shown. There are therefore four valves at each end of the cylinder-bore B^3, and two of these four open inwardly into the bore as suction-valves, while the other two open outwardly from the bore as discharge-valves, a pair of suction-valves being on one side of partition I, and thus between one of the longitudinal compartments and the bore B^3 of the cylinder, and a pair of discharge-valves being on the opposite side of the partition, and thus between the other longitudinal compartment and the bore of the cylinder, it being noted, furthermore, that one compartment of the air-cylinder, which I term the "low-pressure" compartment, because it contains air at the lower of the two unequal pressures already spoken of, is provided with four valves on the ends of the bore, which valves are all suction-valves, and also that the other compartment of the air-cylinder which I term the "high-pressure" compartment, because it contains air at the higher of the two unequal pressures already mentioned, is provided with four valves on the ends of the bore, which valves are all discharge-valves and permit the compressed air to be discharged through them from the bore into the high-pressure compartment; or, as it may be otherwise stated, each end of the bore is furnished with four valves, two suction and two discharge, the suction serving to admit air to the cylinder to be compressed and the discharge to allow the exit of the air after compression, all of said valves opening so as to establish proper communication with the respective compartments. Proceeding now to describe more minutely the specific construction of these valves so as to show one way in which they may be built, it being evident, of course, that the structural details may vary greatly in practice, we see that at each of the openings in the bore ends h' h' a valve-casing i is screwed horizontally into the head. The casings i which belong to the discharge-valves are perforated with a suitable number of openings i' i'. The casings i which belong to the suction-valves are preferably imperforate, but open-ended. In the suction-valves, one of which is shown in detail at the left in Fig. 4, a stationary tubular casting j is supported by a vertical perforated diaphragm within the valve-casing i in such a manner as to leave an annular space between the casting and the casing. k indicates the valve, having an inclined seat and adapted to open inwardly into the bore B^3. The valve-stem k' of valve k lies horizontally within the tubular casting j. On the outer end of stem k' is a nut j', having a flange thereon, and likewise a jam nut or collar j^2, secured in place by a pin. A coiled spring i^2 surrounds the casting j and is tensioned between the flange on nut j' and the diaphragm which supports the casting in position. The spring is so arranged as to normally keep the valve closed; but when the pressure on the other side of the valve in the bore B^3 falls below the pressure in the valve-casing, plus the slight power of the spring, the valve will open, overcoming the resiliency of the spring and compressing the same, and thus permitting air to pass through the valve. Air enters the suction-valve, as will be seen, through the annular opening at the end of the valve-casing. The discharge-valves have a slightly different construction. The casing i of these valves is perforated, as we have seen. A casting l' lies within the perforated casing, so as to leave an annular space, the outer part of this casting being enlarged and screwed into the outer end of the valve-casing. l denotes the valve, having the stem l^2, which lies within a horizontal passage in the casting l', the seat of the valve being beveled and the valve designed to open outwardly from the bore. A spring l^2 is coiled around the casting l', bearing at one end against the valve l and at the other end against a shoulder on said casting. Hence when valve l opens the spring is compressed, and said spring serves to close the valve at the proper time. Obviously when the pressure within bore B^3 against the face of the valve l is greater than the air-pressure within the valve-casing and adjacent compartment, plus the slight power of the spring, the discharge-valve will open and allow the passage of air out of the bore B^3.

From the foregoing description of the construction and location of the suction and discharge valves it will be evident that when the piston B^2 moves to the right in the direction of the arrow as shown in Fig. 4 the two suction-valves in its rear, which lead from the low-pressure compartment of the cylinder, will, in consequence of the vacuum created in the rear of the moving piston, be opened to permit air to flow in behind the piston, said inflowing air being either of atmospheric pressure or of a higher pressure, as the case may be. The two discharge-valves in the rear of the piston, which communicate with the high-pressure compartment of the cylinder, are at this time closed, while the two suction-valves in front of the moving piston will be closed, the air being compressed against them; but the two discharge-valves which lead into the high-pressure chamber in front of the moving piston will, when the air is compressed against them, be opened, allowing the air at a higher pressure to enter the chamber of higher pressure. When the compression-piston reverses its movement, a similar operation will take place, the air now entering through the other two suction-valves, while the air in advance of the piston is delivered in a compressed condition into the high-pressure chamber through the other pair of discharge-valves.

The upper side of the air-cylinder B′ is entered by a vertical pipe K, which discharges air into the channel J, that connects the two head chambers, which, taken together and in conjunction with said channel J, constitute the low-pressure compartment of the air-cylinder. (See Figs. 3 and 4.) The upper side of the air-cylinder is also entered by another vertical pipe G, which receives air from the channel J′, that connects the two head chambers, which, taken together and in conjunction with said passage J′, constitute the high-pressure compartment of air-cylinder B′. The pipe K has a short horizontal branch pipe K′, which is provided with a valve K^2, said branch pipe K′ opening into the atmosphere. The pipe K is therefore an air-induction pipe for delivering air to the cylinder B′ and the pipe G an air-eduction pipe for withdrawing and carrying air from the said cylinder, the former being a "low-pressure" pipe, so called, and conveying air of atmospheric pressure (or of the lower of the two unequal pressures, as the case may be) into the cylinder, while the latter is a "high-pressure" pipe, so called, and conveys compressed air away from the cylinder. When the valve K^2 is open and the apparatus is being charged, the function of pipe K is to deliver atmospheric air to the cylinder; but after said valve has been closed and the valve m has been opened, then the pipe K serves to deliver to the air-cylinder air at a certain lower pressure, but above the atmospheric pressure, while the pipe G serves to carry from the cylinder compressed air at a certain higher pressure than the air in the induction-pipe, the air in the two pipes being thus of unequal pressures. The other air-cylinder B is suitably entered by induction and eduction pipes.

L denotes the induction or low-pressure pipe, situated vertically above cylinder B and entering through the upper side of the cylinder into the channel that forms a part of the low-pressure compartment of the cylinder. M denotes the eduction or high-pressure pipe located vertically above the cylinder B and entering through the upper side of the cylinder into the channel that forms a portion of the high-pressure compartment of this cylinder. The pipe L has a short horizontal branch L′, which is provided with a valve L^2, said branch pipe L′ opening into the atmosphere in the same way as the pipe K′. It will be readily seen that the function of the pipes L and M is the same as that of the pipes K and G. The pipes L and K, having a similar function, (being both low-pressure pipes,) are connected together by the horizontal pipe L^3, and likewise the pipes G and M, having a similar function, (being both high-pressure pipes,) are connected together by the horizontal pipe G′. The pipes K and M, being the main low and high pressure pipes, extend upward vertically for some distance, or as far as may be necessary. They are connected together at one point by a horizontal pipe N, (see Fig. 3,) having therein a valve N′, adapted to be operated by hand, and at a point above pipe N the pipes K and M are connected by another horizontal pipe O, which is provided with an automatic valve O′. The pipe O is only used at certain times, as will be hereinafter related. At the upper end of the low-pressure pipe K a pipe P is connected thereto, which runs from the motor or driven apparatus where the power is applied to the work, and also at the upper end of the high-pressure pipe M a pipe P′ is connected thereto, which likewise leads to the driven apparatus, said pipe P′ conveying the high-pressure air to said apparatus, while the pipe P carries the low pressure or exhaust back from said apparatus and assists in returning it to the compressor. The low-pressure pipe K is provided at a suitable point with a gage o for indicating the pressure in the pipe, and also with a safety-valve p for relieving any undue or dangerous pressure, and likewise the high-pressure pipe M is provided with a gage o' and a safety-valve p'. In the low-pressure pipe K, at a suitable point above where the horizontal pipe L^3 is coupled thereto, is a valve m, the function of which will be stated in describing the operation.

In Fig. 10 I have shown a plan view of the entire apparatus, including a plan view of one form of the driven machine. This driven machine or apparatus may obviously be of any kind and for any purpose or work, such as a common engine or a rock-drill pump or any other machine.

In the example delineated in Fig. 10, V denotes a cylinder, r a piston-rod, and V′ a driving-shaft, while r' is the connecting-rod. The low-pressure pipe P and the high-pressure pipe P′ are both connected to cylinder V, so that low-pressure air is on one side of the piston when high-pressure air is on the other side thereof, leaving the piston unbalanced and permitting the engine to operate as usual. This bare outline of a driven machine will serve to explain sufficiently and indicate how the pipes leading thereto are arranged and the power applied.

In order to govern the air-pressures, keeping them regular and constant and maintaining a fixed ratio to each other, I provide what I call an "air-governor." This governor has two main functions: first, to regulate the speed of the compressor with relation to the work to be done by the driven machine—*i. e.*, to proportion the speed of the compressor so that any amount of work may be done that is required (within, of course, the limit of capacity of the machine)—and, second, to maintain the desired ratio between the pressures in the circulating-pipes.

n and n' designate two pipes leading from opposite ends of the pipe N, or one leading from high-pressure pipe M and one leading from low-pressure pipe K, said pipe n being the one that leads from low-pressure pipe K and is filled with low-pressure air, while said

pipe n' is the one leading from high-pressure pipe M, and is consequently filled with high-pressure air. The course of the pipes n and n' is seen in Figs. 1 and 2.

The construction of the air-governor is seen in detail in Figs. 8 and 9. At the upper end of the governor is a vertical cylinder Q, containing a trunk-piston Q′, having a certain area. This cylinder I preferably term the "low-pressure-governor" cylinder, because it is at that end of the governor where the low-pressure pipe n arrives, and hence low-pressure air acts upon piston Q′. The cylinder Q above the piston Q′ is filled with oil or some lubricant, and the upper end of the cylinder is provided with a projecting pipe n^3, through which the lubricant is fed into the cylinder when needed. This pipe n^3 is entered by the low-pressure pipe n, and therefore, speaking in strictness, the low-pressure air acts against the lubricant instead of against the piston; but the surface is of the same area, and hence the result is the same. Between the upper head of cylinder Q and the piston Q′ is a spring q, immersed in the lubricant, which spring allows the valve to open and close gradually and cushions the movement. Furthermore, it will be noted with respect to cylinder Q that its lower end is provided with an oil-drip pipe r^2 to receive any oil that may possibly leak between the piston and cylinder. At the lower end of the governor is another cylinder R, containing a trunk-piston R′, having a certain area less than the area of piston Q′. This cylinder I preferably term the "high-pressure-governor" cylinder, because it is situated at that end of the governor where the high-pressure pipe n' arrives, and hence high-pressure air acts upon the piston R′. The lower end of the cylinder R is filled with lubricating-liquid inserted thereinto through the bent pipe n^2, the upper end of which is higher than cylinder R. This pipe is entered by the high-pressure air-pipe n', and therefore, speaking more accurately, the pressure of the air is exerted directly upon the oil or lubricant instead of upon the piston R′; but the result is the same. Between the upper cylinder Q and the lower cylinder R is a hollow valve-chamber H^2, through which the steam-pipe H′ passes. Said valve-chamber H^2 is secured to cylinder Q by the vertical bolts r r, passing through flanges on the cylinder and on the chamber. It is also connected to cylinder R by the vertical bolts r' r', which pass through suitable flanges on the valve-casing and on the cylinder R. In this way the two cylinders and the valve-casing are firmly fastened together in such a manner as to leave a convenient space between the valve-casing and each cylinder, the bolts or rods r r and r' r' being long enough to permit of this.

S indicates a yoke having a general oval form and surrounding loosely the valve-chamber H^2, the upper end of the yoke being in suitable proximity to the bottom of the cylinder Q, while the lower end of the yoke is near to the upper end of cylinder R.

T denotes a short bar or rod, the upper end of which enters loosely a socket in the lower end of the piston Q′, said bar passing through the bottom of cylinder Q, while the lower end of the bar or rod T enters loosely a socket in the top end of yoke S. This rod T is held in place by the downward pressure of the piston Q′. Another rod or bar T′ has its upper end received loosely into a socket at the bottom of the yoke S, while its lower end loosely enters a socket in the piston R′, belonging in the high-pressure cylinder R. The rod T′ is held in place by the upward pressure of the piston R′. Near its upper end the yoke S is provided with a cross-connection s, having a nut which holds the screw-threaded end of a valve-rod s'. This rod carries the valve H^3 at its lower end within the steam-pipe H′, said valve being of any suitable and preferred construction and arranged to graduate the flow of steam through the steam-pipe from the boiler to the engine in proportion to the amount of work to be done, and also to control the flow whenever it may be necessary to thus cut off or graduate the amount of steam, and thereby govern the speed of the engine and compressor. Said valve operates automatically. I will now detail the operation of this governor device. It keeps the ratio of the air-pressures constant and controls the speed of the compressor in proportion to the amount of work to be performed. Its function is not to maintain a uniform speed under all circumstances, as is the case with the ordinary steam-engine governor; but my air-governor is designed to automatically vary the speed of the compressor to suit the work being done by the driven machine, and the speed of the compressor is of course by varying the speed of the motor-machine, which in the present example is the steam-engine. Thus one kind of driven device—such as one rock-drill—may require a certain number of revolutions of the steam-engine per minute to actuate the compressor sufficiently to transmit the required power to drive said drill. Obviously two of these drills will require twice as many revolutions of the actuating-engine to drive them. The air-governor automatically varies the steam-supply so as to control the speed of the engine and enable it to perform the required duty. Likewise the said governer keeps the ratio of pressure constant.

If at any time the compressor runs faster than is necessary, so as to compress more air than is required to drive the driven machine, the pressure in the high-pressure pipe will increase and that in the low-pressure pipe will diminish, and the variation of pressures influences the air-governor, so that it tends to lessen the speed of the compressor and restore the normal ratio of pressures. On the other hand, if at any time the compressor is running slower than is necessary to compress air as fast as the driven device requires, the press-

ure in the high-pressure pipe will decrease and that in the low-pressure increase, and this variation of the ratio, acting on the air-governor, tends to increase the speed of the compressor and restore the normal ratio of pressures. When the same conditions of work to be done obtain, the air-governor keeps the ratio of pressures constant by regulating the speed of the motive power and keeping it constant so long as the conditions of work do not change. When the conditions of work vary through inequalities in the rock or a multiplication of the drilling devices, (supposing the driven device to be a drill,) then the air-governor operates upon the motor-engine and automatically changes and proportions its speed to conform to the change in the work and make it capable of doing more or less work, as may be required, maintaining all the while a constant work. Evidently increase of work to be done will lessen the amount of high-pressure air, thus causing the valve to open wider and the engine to run faster to supply the required amount of high-pressure air.

The areas of the two pistons Q′ and R′ have the same ratio as the air-pressures. Thus, if the high-pressure air is two hundred pounds to the square inch and the low-pressure air is one hundred pounds to the square inch, the area of piston Q′ will be twice the area of piston R′, since the pressures upon one inch of surface of piston R′ will be balanced by the pressure upon two inches of surface of piston Q′. In this case, therefore, when the pressure in pipe *n* is one hundred pounds and the pressure in pipe *n′* is two hundred pounds, these pressures will exactly balance each other in the governor and the pistons and other parts will remain in equilibrium. When the governor is in this condition, the parts may be adjusted up or down by hand and (but for the spring *q*) would remain in any position where they might be placed. The spring *q*, however, is interposed above the piston Q′, so as to add a little excess of pressure to that end of the governor and disturb what would otherwise be an equilibrium sufficiently to keep the valve H^3 normally open, although the spring *q* is of slight power and readily overcome by any slight increase in the high-pressure air. The spring is further advantageous in allowing a gentle motion in the parts of the governor. It cushions the end of the piston and prevents the shock which might take place when a change of ratio suddenly occurs and disturbs the equilibrium. The present example of governor is described simply as an example. Especially is this true with respect to the areas of the pistons Q′ and R.′ These areas vary, of course, in different governors where different ratios between the air-pressures than that of two to one are employed in the apparatus.

In order to change the ratio of pressures without altering the pistons, weights may be hung upon the yoke S at the perforations S′.

I will now explain the operation of the main apparatus.

Referring to Figs. 2 and 3, I will first describe the operation when cross-pipe N, having valve N′, is used and cross-pipe O is supposed to lie idle or is removed from the apparatus. First, the valves K^2 and L^2 will be opened to admit atmospheric pressure air into the pipes L′ and K′ and thence into the compressing-cylinders. The valve N′ will also be opened, usually by the hand of the operator, it being non-automatic, and the valve *m* closed. Then the compressor will be set to work, and air will be drawn from the atmosphere into the compressing-cylinders, compressed therein, and sent through the pipes G M N K above valve *m* and pipes P and P′. The compressed air in the entire system will of course be now at the same pressure and a like degree of pressure will be indicated on both gages—that belonging to the high-pressure pipe and that belonging to the low-pressure pipe. The operation of compressing will be continued until both gages register, say, one hundred pounds, supposing this to be the amount of the lower of the two unequal pressures. Then the operator will close valve N′. This will separate the system of low-pressure pipes from the system of high-pressure pipes, leaving the pressure in the former system fixed permanently at one hundred pounds. The operation of the compressor will continue until the air in the high-pressure pipes has attained a pressure of two hundred pounds, when the inlet-valves K^2 and L^2 will be closed by hand, preventing the admission of any more air, and the valve *m* will be opened by hand, allowing air at a pressure of one hundred pounds to be delivered to the compressor-cylinders. We have now our closed system, wherein the air circulates between the compressor and the motor. The air in one part of the apparatus is at one hundred pounds pressure, as will be seen from one of the gages, and the air in the other part of the apparatus is at two hundred pounds pressure, as will be seen on the other gage. Here, then, are the two unequal pressures of air, both above the atmospheric pressure, the ratio of the pressures being as two to one, which ratio is kept constant by means of the air-governor, and this governor is found in actual practice to do its work so perfectly that hardly any fluctuation of the ratio is perceptible upon the gages or indicators with which the apparatus is provided. It will be noted that no atmospheric pressure air is drawn into the machine during its operation. The compressor always works upon air at one hundred pounds pressure instead of upon air at atmospheric pressure. To supply any trifling leakage that may take place, I provide a little air-pump U, (see Fig. 1,) which may be of any suitable and ordinary construction and which delivers into the low-pressure pipes at some suitable point.

I will now briefly describe the operation when the cross-pipe O, having the automatic valve O', is made use of. In this case the pipe N serves no necessary purpose, although the valve N' might be left open until all the pipes are filled with air at one hundred pounds pressure, but lies idle with the valve N' closed, and also the air-governor is not used, but the valves in the air-pipes n and n', running to the governor, (said valves are shown in Fig. 1,) are closed. In starting the apparatus the valves K^2 and L^2 will be open and the valve m closed. The valve O' will likewise be closed. This valve is constructed to operate automatically, and is so arranged that it will open only when the pressure on one side of it is one hundred pounds greater than the pressure on the other side. Of course it may be adjusted for other pressures; but in the present example we suppose it to open at one hundred pounds pressure on one side above that on the other.

The detailed construction of the automatic valve is shown in section in Fig. 11. It consists simply of any ordinary valve w. W is the screw-threaded valve-rod having thereon a nut W^2, tongued to slide in grooves in the side of the valve-casing, the upper end of the valve-rod W projecting through the top of the casing and formed for the attachment thereto of a wrench, whereby the rod may be rotated and the nut W^2 moved up or down. A spring w' envelops rod W and bears at one end against nut W^2 and at the other against the valve w. Thus it is evident that the adjustment of the nut W^2 adjusts the tension of the spring and so regulates the power which holds the valve closed. Hence the valve may be adjusted so as to open at any desired degree of pressure. On the upper part of the rod W, within the casing, is a beveled collar W^3, forming a seat at the upper end of the casing where the rod passes therethrough, so as to provide an air-tight joint at this point, said beveled seat being preferably used in lieu of an ordinary stuffing-box.

As the operation of compression proceeds, the high-pressure pipes will be filled with air having a pressure of one hundred pounds before any air will enter the low-pressure pipes. As soon as the pressure against the sides of valve O' attains one hundred pounds the valve will open and permit air to pass into the low-pressure pipes. As the compression continues, the air in the low-pressure pipes will finally stand at one hundred pounds and the air in the high-pressure pipes at two hundred pounds. The compression may continue beyond this, if desired, but probably it will not be. Then the valves K^2 and L^2 will be closed and the valve m opened, and so we again have our closed system of pipes, within which the air circulates constantly, one gage standing at one hundred pounds and the other at two hundred pounds when the the ratio is one to two. This automatic-valve arrangement which I have just described is used only in certain cases, chiefly when the compressor is driven by a belt from some motor-shaft which actuates other machinery, which machinery it is necessary to drive with a steady uniform motion, and so the motor-shaft cannot have its speed varied, and hence an air-governor cannot be used, and therefore the automatic valve serves the purpose and keeps the difference of the pressures constant. It will be noted that this arrangement with pipe O, having valve O', accomplishes a different result from that accomplished by the air-governor, in that, while the air-governor serves to keep the ratio of the two unequal air-pressures constant, this automatically-operating valve serves simply to keep the difference of the pressures constant. In other words, it may be said that the automatic valve serves to regulate the arithmetical ratio of the two pressures, while the air-governor serves to keep the geometrical ratio of the two pressures constant. This is an important difference when we come to look at the result that is accomplished. Although this arrangement with pipe O is generally used with belt-machines and not in connection with the first arrangement, yet it may be made use of in connection therewith if at any time the air-governor should become disabled, and I make no special claim to novelty in the arrangement of the pipe O, with its regulating-valve, but simply include the same herein to more fully explain the uses and adaptation of my improved machine.

Numerous advantages accrue from the use of the invention herein described, a few of which it may be well to enumerate here, in order that the importance of the invention and the great utility and advantage in employing two unequal air-pressures, both above the atmospheric, may be clearly understood and appreciated.

Among the resulting benefits I will first mention that my machine for transmitting power by means of compressed air is much smaller and more compact, and therefore cheaper, easier of transportation, and very much lighter in weight than the machines now in use for doing a corresponding amount of work. This statement is easily proved by a simple calculation. Suppose it is desired to provide a cubic foot of air having a pressure of one hundred pounds at one stroke of the piston. Obviously if air at atmospheric or fifteen pounds pressure is to be compressed to this required pressure, the original volume of atmospheric air will be seven cubic feet or thereabout—*i. e.*, seven times the volume of the required volume at one hundred pounds pressure. If air at fifty pounds pressure is to be compressed to the pressure of one hundred pounds, two cubic feet of the former will be required to make one of the latter.

In the ordinary air-compressor air is received into the machine at atmospheric pressure, and hence the cylinder must contain seven cubic feet, in order that the atmos-

pheric air may at one piston-stroke be compressed to a cubic foot of one hundred pounds pressure. Suppose it is desired to work the ordinary air-compressor with an unbalanced pressure of one hundred pounds. In order that it may do this, air must be compressed to one hundred and fifteen pounds, so that there may be one hundred and fifteen pounds on one side of the piston and fifteen pounds, or atmospheric pressure, on the other side of it.

In my apparatus after it is charged, the air is delivered to the compressor at one hundred pounds pressure. Therefore if it is desired to obtain a cubic foot of air having two hundred pounds pressure, which in my machine will give an unbalanced pressure of one hundred pounds to work with, the cylinder needs to contain only two cubic feet of one hundred pounds air. Hence where a cylinder in my apparatus required a cubical contents of two cubic feet the cylinder in the ordinary compressor apparatus requires a cubical contents of seven or eight cubic feet in order to effect the same compression at one stroke. This makes manifest the essential difference in respect of size between the two machines, said difference being dependent upon the fact that in one case the compressor operates upon atmospheric pressure air, while in my compressor it acts upon air having already a very high pressure, and this therefore demonstrates the value of my invention in making a small compact machine which is found in practice to have less than half the weight of certain other popular machines doing a similar work.

Another advantage accruing directly from my invention is an increase in efficiency. This may be proved by another simple calculation, as follows: When a volume of air is compressed to one-half, a certain amount of the energy generated is lost through the heating of the air and the consequent dissipation of energy in the form of heat. When a volume of air is compressed to one-third, a certain greater amount of power is lost through heat. When, therefore, a volume of air is compressed to one-seventh, as is the case when atmospheric pressure air is converted into air having a pressure of one hundred and fifteen pounds to the square inch, so as to give an unbalanced pressure of one hundred pounds, a very much greater amount of power is lost through the dissipation of heat energy than is the case when the air is compressed to half its volume, as in my machine, where air at a hundred pounds pressure is compressed to two hundred pounds, so as to give the same unbalanced pressure of one hundred pounds. Therefore it will be seen that I lose very little power from heat, and hence increase the efficiency of the machine very greatly. It is found in actual practice, which also is confirmatory of theoretical calculations on the subject, that a very much greater per cent. of the power generated at one end of the machine is applied to the work at the other end of my machine than is the case in any other with which I am familiar.

Other advantages belonging to my invention might be referred to, but it is deemed unnecessary, as the two which have just been described are sufficient to indicate the importance of my improvement.

Numerous details in the construction and relative arrangement of the parts of the machine herein described may be made without departing from the invention, and I reserve the liberty of varying the same as experience may suggest within wide limits.

Having thus described my invention, what I claim as new, and desire to secure by Letters Patent, is—

1. In an apparatus for transmitting power by means of compressed air or other gas circulating in a closed system at two unequal pressures, both above normal atmospheric pressure, the combination of a compressor, a machine to be driven by the power thereby generated, an air-conduit between the compressor and the driven machine containing air at a certain lower pressure and another air-conduit between the compressor and the driven machine containing air at a certain higher pressure, and means for keeping the geometric ratio of the two pressures constant.

2. In an apparatus for transmitting power by means of compressed air or other gas or fluid circulating in a closed system at two unequal pressures, both above normal atmospheric pressure, the combination of a compressor, a machine to be driven thereby, two air-conduits between the compressor and the machine containing air at the unequal pressures, and a governor for keeping the geometrical ratio of the two pressures constant and regulating the speed of the compressor in proportion to the work to be done by the driven machine.

3. In an apparatus for transmitting power by means of compressed air or other gas circulating in a closed system at two unequal pressures, both above normal atmospheric pressure, the combination, with a compressor, a driven machine, and air-conduits between the compressor and the driven machine, of an inlet-valve for admitting atmospheric air to the compressor in charging, a valve in the low-pressure conduit adapted to be closed in charging, and a connection between the two conduits provided with a valve adapted to be operated by hand to make a temporary communication between said conduits during the operation of charging.

4. In an apparatus for transmitting power by means of compressed air or other gas circulating in a closed system at two unequal pressures, both above normal atmospheric pressure, the combination of a compressor, a driven machine, air-conduits between the compressor and driven machine, an inlet-valve for admitting atmospheric air to the compressor in charging, a valve in the low-pressure conduit adapted to be closed in

charging, a connection between the two conduits provided with a suitable hand-valve, whereby the operator in charging may temporarily establish communication between the conduits, and an air-governor for keeping the ratio of the two pressures constant and for regulating the speed of the compressor in proportion to the amount of work to be performed by the driven machine, together with suitable connections between the air-conduits and the governor.

5. In an apparatus for transmitting power by means of compressed air or other gas circulating in a closed system at two unequal pressures, both above normal atmospheric pressure, the combination of the compressor, a motor for actuating it, a driven machine, two air-conduits between the compressor and driven machine containing air at two unequal pressures, and a governor for keeping the geometric ratio between the two pressures constant by regulating the speed of the motor which drives the compressor and also for regulating the speed of the compressor in proportion to the amount of work to be performed by the driven machine.

6. In an apparatus for transmitting power by means of compressed air circulating in a closed system at two unequal pressures, both above normal atmospheric pressure, the combination of a compressor, a suitable steam-engine for actuating the same, a motor or apparatus to be driven by the compressed air, two air conduits or channels between the compressor and this machine containing air at unequal pressures, and an air-governor for keeping the geometric ratio between these two pressures constant, said governor controlling a valve in the steam-supply pipe of the engine and thus regulating the speed of the compressor so as to keep the ratio of pressures constant, said governor having also the function of proportioning the speed of the compressor to the amount of work to be performed, substantially as herein described.

7. In an apparatus for transmitting power by means of compressed air or other gas circulating in a closed system at two unequal pressures, both above normal atmospheric pressure, the combination of a compressor, a machine to be driven by the compressed air, an air-conduit between the compressor and the driven machine adapted to contain air at the lower of the pressures, another air-conduit between the compressor and the driven machine adapted to contain air at the higher of the two pressures, a suitable motor for actuating the air-compressor, an inlet-valve for admitting atmospheric air to the compressor in charging, a valve in the low-pressure conduit adapted to be closed in charging, a connection between the two conduits provided with a hand-valve, whereby the operator during the process of charging up the apparatus may temporarily establish a communication between said conduits, an air-governor containing two pistons whose areas have a ratio equal to the ratio between the two air-pressures, said governor controlling the speed of the motor which actuates the compressor and discharging the double function of maintaining a constant geometric ratio between the two air-pressures and also proportioning the speed of the compressor to the amount of work to be performed by the driven machine, a low-pressure air-pipe running from the low-pressure conduit to the low-pressure end of the governor, and a high-pressure pipe running from the high-pressure conduit to the high-pressure end of the governor.

8. In an apparatus for transmitting power by means of compressed air circulating in a closed system at two unequal pressures, both above the normal atmospheric pressure, the combination of a compressor, a suitable motor for actuating it, a machine to be driven by the power of the compressed air, two air-conduits between the compressor and the driven machine containing air at unequal pressures, an air-governor having two pistons whose areas have the same ratio as that between the unequal air-pressures and having also a valve-chamber containing a valve which controls the supply-pipe leading to the motor which actuates the compressor, said valve being regulated by the movement of the governor-pistons, a low-pressure pipe running from the low-pressure conduit to the low-pressure end of the governor, and a high-pressure pipe running from the high-pressure end of the conduit to the high-pressure end of the governor, so that the governor-pistons may be balanced between the two pressures.

9. In an apparatus for transmitting power by means of compressed air or other gas circulating in a closed system at two unequal pressures, both above normal atmospheric pressure, the combination of a compressor having its cylinder or cylinders provided with a low-pressure compartment and a high-pressure compartment, the low-pressure compartment being provided with suction-valves which admit to the bore of the cylinder and the high-pressure compartment being provided with discharge-valves which lead from the bore of the cylinder, an inlet-valve for admitting atmospheric air to the compressor in charging, a driven machine, an air-conduit between the compressor and the driven machine which contains air at the lower of the pressures, another air-conduit between the compressor and the machine which contains air at the higher of the pressures, a valve in the low-pressure conduit adapted to be closed in charging, and a connection between the two conduits provided with a hand-valve, whereby the operator during the process of charging up the apparatus may temporarily establish a communication between the two conduits.

10. In an apparatus for transmitting power by means of compressed air or other gas circulating in a closed system at two unequal pressures, both above normal atmospheric pressure, the combination of a compressor having its

cylinder or cylinders provided with a low-pressure compartment having suction-valves leading into each end of the cylinder-bore, and provided also with a high-pressure compartment having discharge-valves leading from each end of the cylinder-bore, a low-pressure pipe entering the low-pressure compartment of the cylinder and running to the driven machine, said pipe being provided with an inlet-valve to admit atmospheric air to the cylinder and also with a valve at a suitable distance from the cylinder, which is closed in charging, and a high-pressure pipe entering the high-pressure compartment of the cylinder and running to the driven machine, together with a connection between said pipes provided with a suitable valve, whereby the operator may during the operation of charging up the apparatus temporarily establish a communication between the two conduits, and a governor for keeping the ratio of the two unequal air-pressures constant by regulating the speed of the motor which drives the compressor and for proportioning the speed of the compressor to the amount of work to be performed by the driven machine.

11. In an apparatus for transmitting power by means of compressed air, the combination of a compressor having its cylinder or cylinders provided with two longitudinal compartments, one of which is a low-pressure compartment and the other a high-pressure compartment, the low-pressure compartment having suction-valves at each end of the cylinder-bore and the high-pressure compartment having discharge-valves at each end of the cylinder-bore, a driven machine, two air-conduits between the compressor and the machine containing air at the two unequal pressures, and the governor for keeping the pressures at a constant geometric ratio by regulating the speed of the motor which actuates the compressor and for proportioning the speed of the motor to the amount of work to be performed by the driven machine.

12. In an apparatus for transmitting power by means of compressed air, the combination of a compressor, a driven machine, an air-conduit connected thereto and containing the air at the lower pressure, another air-conduit containing the air at the higher pressure, a governor device for keeping the ratio of these two unequal air-pressures constant and for proportioning the speed of the compressor to the amount of work to be performed by the driven machine, together with an inlet-valve for admitting atmospheric air to the compressor in charging, a valve in the low-pressure conduit adapted to be closed in charging, and a suitable hand-valve in a connection between the two conduits, whereby the operator during the process of charging the apparatus may temporarily establish a communication between the conduits.

13. In a machine for transmitting power by means of compressed air, a compressor consisting of a cylinder or cylinders having hollow heads, a water-jacket surrounding the cylinder-bore, two longitudinal compartments within the cylinder, one adapted to contain low-pressure air and the other high-pressure air, both pressures being above normal atmospheric pressure, one compartment being entered by a low-pressure pipe and the other by a high-pressure pipe, respectively, which pipes run to a driven machine and are connected by a pipe provided with a suitable valve, which is temporarily opened by the operator during the process of charging up the apparatus, and the suction and discharge valves at each end of the cylinder-bore.

14. In a machine for transmitting power by means of compressed air, the herein-described compressor consisting of a cylinder or cylinders having the partitioned heads whose chambers are connected by channels to form two compartments in the cylinder, a high-pressure and a low-pressure compartment, together with the suction and discharge valves at each end of the cylinder-bore, the suction-valves entering the low-pressure compartment and the discharge-valves entering the high-pressure compartment, said suction-valves consisting, essentially, of the cylindrical casing, the casting for supporting the valve-stem and the spring, said suction-valves opening inwardly into the cylinder-bore, and said discharge-valves consisting, essentially, of the cylindrical perforated casings, the castings supporting the valve-stem and the spring, and said discharge-valves opening outwardly from the cylinder-bore, substantially as described.

15. In a machine for transmitting power by means of compressed air, the combination of the compressor having a cylinder or cylinders provided with the partitioned heads, the channels communicating between the chambers of said heads to form a low-pressure and a high-pressure compartment within the cylinder and the discharge and suction valves at each end of the cylinder-bore, said suction-valves belonging to the low-pressure compartment and said discharge-valves belonging to the high-pressure compartment, said suction-valves having a casing, a casting supporting the valve-stem and a spring, and said discharge-valves having a casing, a casting supporting the valve-stem and a spring, the suction-valves opening inwardly into the cylinder-bore and the discharge-valves opening outwardly therefrom, a driven machine, and an air-conduit running from the low-pressure compartment to said machine and the air-conduit running from the high-pressure compartment to said machine, the air in the two conduits being of unequal pressures and maintained at a fixed geometric ratio by a governor having the double function of keeping these pressures constant and of proportioning the speed of the compressor to the amount of work to be performed by the driven machine.

16. In a machine for transmitting power by means of compressed air, the combination of a compressor consisting of a cylinder or cylinders having two longitudinal compartments, a high-pressure and a low-pressure, the suction and discharge valves at each end of the cylinder-bore, said suction-valves opening inwardly into the bore from the low-pressure compartment and said discharge-valves opening outwardly from the bore into the high-pressure compartment, a machine driven by the compressed air, an air-conduit between the compressor and the machine containing air at a certain lower pressure, another air-conduit between the compressor and the machine containing air at a certain higher pressure, suitable valves for use in charging the apparatus and creating the two unequal pressures, and a device for keeping the ratio of the two pressures constant and for proportioning the speed of the compressor to the amount of work to be performed by the driven machine.

17. In a machine for transmitting power by means of compressed air, the combination of the duplex compressor having two cylinders, each provided with two compartments, a high-pressure and a low-pressure, and suction and discharge valves on each end of the cylinder-bore, the discharge-valves opening from the bore into the high-pressure compartment and the suction-valves opening into the bore from the low-pressure compartment, low-pressure pipes entering the low-pressure compartment of each cylinder, said pipes being connected together, high-pressure pipes entering the high-pressure compartment of each cylinder, said high-pressure pipes being connected together, one of said high-pressure pipes being the main high-pressure pipe and running to the driven machine and one of the low-pressure pipes being the main low-pressure pipe and running to the driven machine, a valve in the main low-pressure pipe, and the connection between the main low-pressure and main high-pressure pipes provided with a suitable valve adapted to be temporarily opened by the operator during the process of charging up the apparatus, so as to establish a communication between the low-pressure and high-pressure air-pipes, substantially as described.

18. In a machine for transmitting power by means of compressed air, the combination of the duplex compressor having two cylinders, the engine for actuating the said compressor, low-pressure pipes entering the low-pressure compartment of said cylinders, each of said low-pressure pipes being provided with an inlet-valve to admit atmospheric air in charging up the apparatus, the high-pressure pipes entering the high-pressure compartment of each cylinder, one of the high-pressure pipes being the main high-pressure pipe and one of the low-pressure pipes being the main low-pressure pipe, a valve in the main low-pressure pipe, a driven machine which is entered by the main low-pressure and main high-pressure pipes, a connection between these two pipes provided with a valve, and a governor for keeping the ratio between the two unequal air-pressures constant at all times.

19. In a machine for transmitting power by means of compressed air, the combination of a compressor having a pair of cylinders, a steam-engine for actuating this compressor, a driven machine adapted to be operated by compressed air, the low-pressure pipe K, entering the low-pressure compartment of one cylinder, provided with an inlet-valve to admit atmospheric air in charging, provided also with another valve and running to the driven machine, the low-pressure pipe L, entering the low-pressure compartment of the other cylinder, provided with an inlet-valve for admitting atmospheric air and connected with the low-pressure pipe K by the pipe L^3, the high-pressure pipe M, entering the high-pressure compartment of one cylinder and running to the driven machine, the high-pressure pipe G, entering the high-pressure compartment of the other cylinder and connected with pipe M by the pipe G', and the connection N between pipes K and M, provided with valve N', substantially as described.

20. The combination of the compressor having two cylinders, the driven machine, the low-pressure pipe K, having a branch pipe K', provided with an inlet-valve K^2 and having also a valve m, the low-pressure pipe L, entering the other cylinder and having a branch pipe L', provided with an inlet-valve L^2, said pipe L connected to the pipe K by the pipe L^3, the high-pressure pipe M, entering one cylinder and running to the driven machine, the high-pressure pipe G, entering the other cylinder and connected with pipe M by pipe G', and the pipe N between pipes K and M, provided with valve N'.

21. In a machine for transmitting power by means of compressed air, the combination of the compressor having two cylinders B and B', the motor for actuating this compressor, the driven machine, the low-pressure pipe K, entering cylinder B', having branch pipe K', with inlet-valve K^2, and having also valve m and gage o, the low-pressure pipe L, entering the cylinder B and having branch pipe L', with valve L^2, said pipe L connecting with pipe K by pipe L^3, the high-pressure pipe G, entering cylinder B', the high-pressure pipe M, entering cylinder B and connected with pipe G by pipe G', said pipes K and M running to the driven machine and connected by pipe N, having valve N', together with the governor for keeping the unequal air-pressures at a constant ratio, and the pipes n and n', running to said governor.

In testimony whereof I affix my signature in presence of two witnesses.

CHARLES CUMMINGS.

Witnesses:

WM. L. BOYDEN,

GEO. L. CLARK.

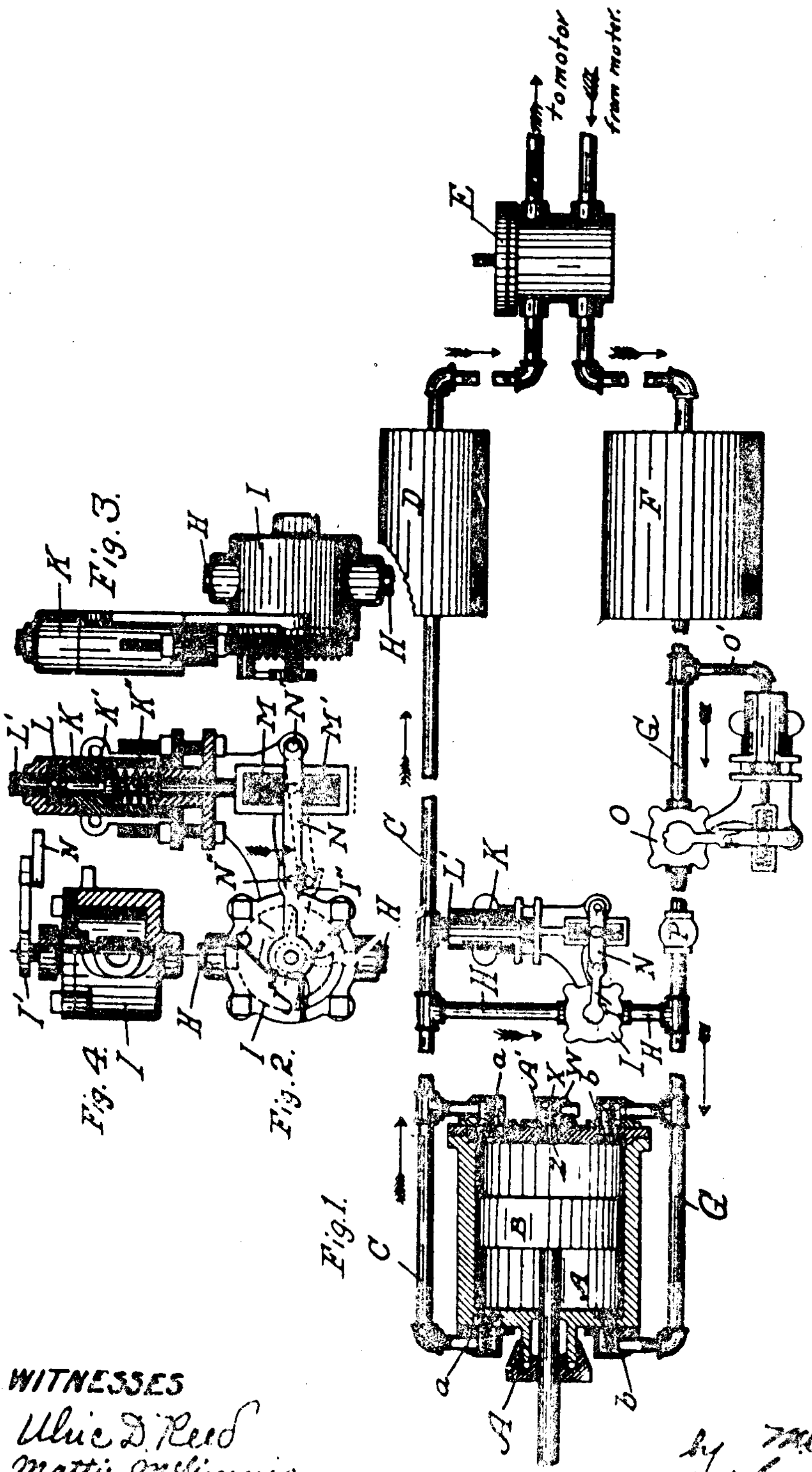
No. 767,027. PATENTED AUG. 9, 1904.

M. C. WILKINSON.

AIR COMPRESSION AND UTILIZING DEVICE.

APPLICATION FILED DEC. 13, 1900.

NO MODEL.



WITNESSES

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No. 767,027. Patented August 9, 1904.

UNITED STATES PATENT OFFICE.

MELVILLE C. WILKINSON, OF LOS ANGELES, CALIFORNIA.

AIR COMPRESSION AND UTILIZING DEVICE.

SPECIFICATION forming part of Letters Patent No. 767,027, dated August 9, 1904.

Application filed December 13, 1900. Serial No. 39,778. (No model.)

To all whom it may concern:

Be it known that I, MELVILLE C. WILKINSON, a citizen of the United States, residing at Los Angeles, county of Los Angeles, State of California, have invented new and useful Improvements in Apparatus for Recompressing, Conveying, and Distributing Compressed Air as a Motive Force, of which the following is a specification.

My invention relates to certain novel modes of controlling the compression, recompression, conveyance, and distribution of air under pressure in that class of machinery in which compressed air is used as a means to transmit motive force to drive machinery more or less distant from the primary source of power in which the exhausted air is recompressed for reuse.

The objects of my invention are to reduce the size and weight of compressed-air machinery, to do away with water-jacketing or other cooling devices, no matter what pressure is used, to enable unevenly-developed power to be utilized at any speed or in any direction, and to generate only the power required to operate the working machinery. I accomplish these objects by means of the mechanism described herein, and illustrated in the accompanying drawings, in which

Figure 1 is a plan view of my improved apparatus, partly in horizontal section, the connecting-pipes being broken away in places. Fig. 2 is a plan view of the quick-acting valve I and its operating mechanism, the piston-chamber being shown in central longitudinal section. Fig. 3 is a side view of the valve I and connections. Fig. 4 is an edge view of the valve I, one-half being in section.

A is an air-compressing cylinder of ordinary construction without cooling devices and is provided with a third or auxiliary ordinary air-admission port A′ at one end of the cylinder only. Upon the oscillation of the piston B air is forced alternately from the space on the sides of piston B through the emission-valves *a a* into pipe C, and from thence into reservoir D, and thence into controller E, and thence to the working machinery. On the return from the working machinery the air again passes through the controller into reservoir F and thence through pipe G and admission-valves *b b* into the air-compressor. The pipes C and G are connected by a by-pass pipe H, on which is placed a quick-acting valve I and its operating mechanism, (hereinafter more fully explained,) which when the pressure in pipe C reaches a predetermined point—say two hundred pounds—is automatically opened by the mechanism shown in section in Fig. 2 and consists of the chamber K, containing a piston K′, held against the air-pressure in chamber L by spring K″. Chamber L is connected by pipe L′ to pipe C. When the pressure in pipe C reaches one hundred and ninety-five pounds, the piston K begins to compress spring K″ and also spring M. When the tension of spring M has reached, for example, five pounds, this pressure being communicated to lever N, pivoted at N′, and pin N″ within fork of lever I″ of valve I, the expansion of spring M will move lever N in the direction of the arrow, causing the mechanism to assume the position shown in dotted lines in Fig. 2, with the spring J then in notch J″ on the hub of the lever I″. The valve is then open. When the pressure in chamber L again returns to one hundred and ninety-five pounds, spring M′ causes spring J to leave notch J″. The mechanism then assumes its first position, and the valve is closed. As long as no air is being used by the working machinery this by-pass H remains open, the air circulating through pipes C H G and compressor A at the same pressure. When air is used by the working machinery, the pressure in pipe C falls, thus closing valve I, when the air is then pumped from pipe G into pipe C. In pipe G is placed quick-acting valve O, similar in construction to valve I and operated by similar mechanism, only the chamber back of the spring-pressed piston is connected to pipe G instead of pipe C by pipe O′. Between the valve O and the by-pass H on pipe G is placed a check-valve P to permit air to pass from the controller E to the compressor A, but will prevent air from passing in the opposite direction from pipe C through by-pass H and valve O to the controller E when valve I is open. Valve O being at one hundred pounds pressure will give one hundred

pounds pressure on the working machinery and allow the compressed air from the compressor to expand to but two volumes, and thus prevent freezing, and as the compressor will then receive air at one hundred pounds pressure it has to compress it to two hundred pounds or to but one-half its volume. Therefore the compressor requires no cooling devices. Until reservoir F has received air to raise its pressure to one hundred pounds and open valve O air is drawn into the mechanism through valve A′ until both reservoirs D and F have been pumped up to pressure, when valve O opens and the compressor pumps direct from reservoir F, which receives the exhaust of the working machinery. Whenever the pressure drops in pipes C and G, due to loss by leakage, and valves I and O are neither of them open, then the compressor pumps what air remains in the pipes G and H between the valves I and O and suction-ports *b b* into pipe C, and when the air in this section has been exhausted below atmospheric pressure then the outside atmospheric pressure causes air to enter at port A′ and be pumped into pipe C.

The reason why it is desirable to have the valve A′ at one end of the cylinder only is that it requires less power to pump the air from atmospheric pressure to two hundred pounds by using the compressor as a single-acting pump than if it were double-acting, for if it were required to pump as a double-acting compressor from atmospheric to two hundred pounds pressure it would require nearly twice as powerful initial motive force as is required after the pipes are charged to one hundred pounds when the compressor has to compress it to but another one hundred pounds.

Having described my invention, what I claim as new, and desire to secure by Letters Patent, is

1. In an air-compression system, the combination with a compression-cylinder provided with an induction and an eduction port at either end thereof, a high-pressure reservoir connected with the eduction-ports, a low-pressure reservoir connected with the induction-ports, a controller with which the reservoirs are connected, and a pipe extending between the connections of the reservoirs and cylinder, of a pressure-operated valve in the pipe, a check-valve in the connection leading from the low-pressure reservoir to the cylinder and a quick-acting valve located in said connection, pneumatic mechanism for operating the valve, the pneumatic mechanism set to operate only when a predetermined pressure is reached.

2. The combination with a compression-cylinder, high and low pressure reservoirs connected therewith, a pipe extending between the connections, a valve in the pipe and means for controlling the valve, of a rotary valve in the connection between the low-pressure reservoir and the cylinder, a chamber connected with the connection, a piston in the chamber, a valve-fork on the valve, a lever connected therewith, the piston adapted to engage the lever and means for normally retaining the piston at one limit of its movement until overcome by a predetermined pressure.

3. In a system for compressing using and recompressing air under pressure as a motive power, a double-acting air-compressor having induction-ports connecting the said compressor with the low-pressure side of the system and eduction-ports connecting the said compressor with the high-pressure side of the system and an induction-port connecting one end only of the compressor with the external air; an inwardly-opening valve in the induction-port leading to the open air, adapted to open and feed air into one end of the compressor only when the pressure therein falls below the external air-pressure.

4. The combination with a compression-cylinder, high and low pressure reservoirs, and pipes connecting the reservoirs, and cylinder, a by-pass pipe extending between the connecting-pipes, of a valve therein, means connected with the valve and with the high-pressure reservoir to operate the valve, the means comprising a chambered casing in communication with the high-pressure reservoir, a plunger in the casing, a spring for retaining the plunger against the normal pressure of the reservoir, a fork-arm connected with the valve, a lever stationarily pivoted at one end the opposite end connected with the fork-arm and a block yieldingly held by the plunger and connected with the lever.

5. The combination with a compression-cylinder, high and low pressure reservoirs connected therewith and a pipe extending between the high and low pressure connections, of a rotary valve in the pipe, a forked arm connected therewith, a chambered casing connected with the high-pressure reservoir, a plunger in the casing, the plunger provided with an extension protruding outside the casing, a block yieldably supported in the plunger extension means for retaining the plunger in position against the normal high pressure, and a lever connected with the arm and with the yieldingly-supported block.

In witness that I claim the foregoing I have hereunto subscribed my name this 30th day of November, 1900.

MELVILLE C. WILKINSON.

Witnesses:

G. E. HARPHAM,

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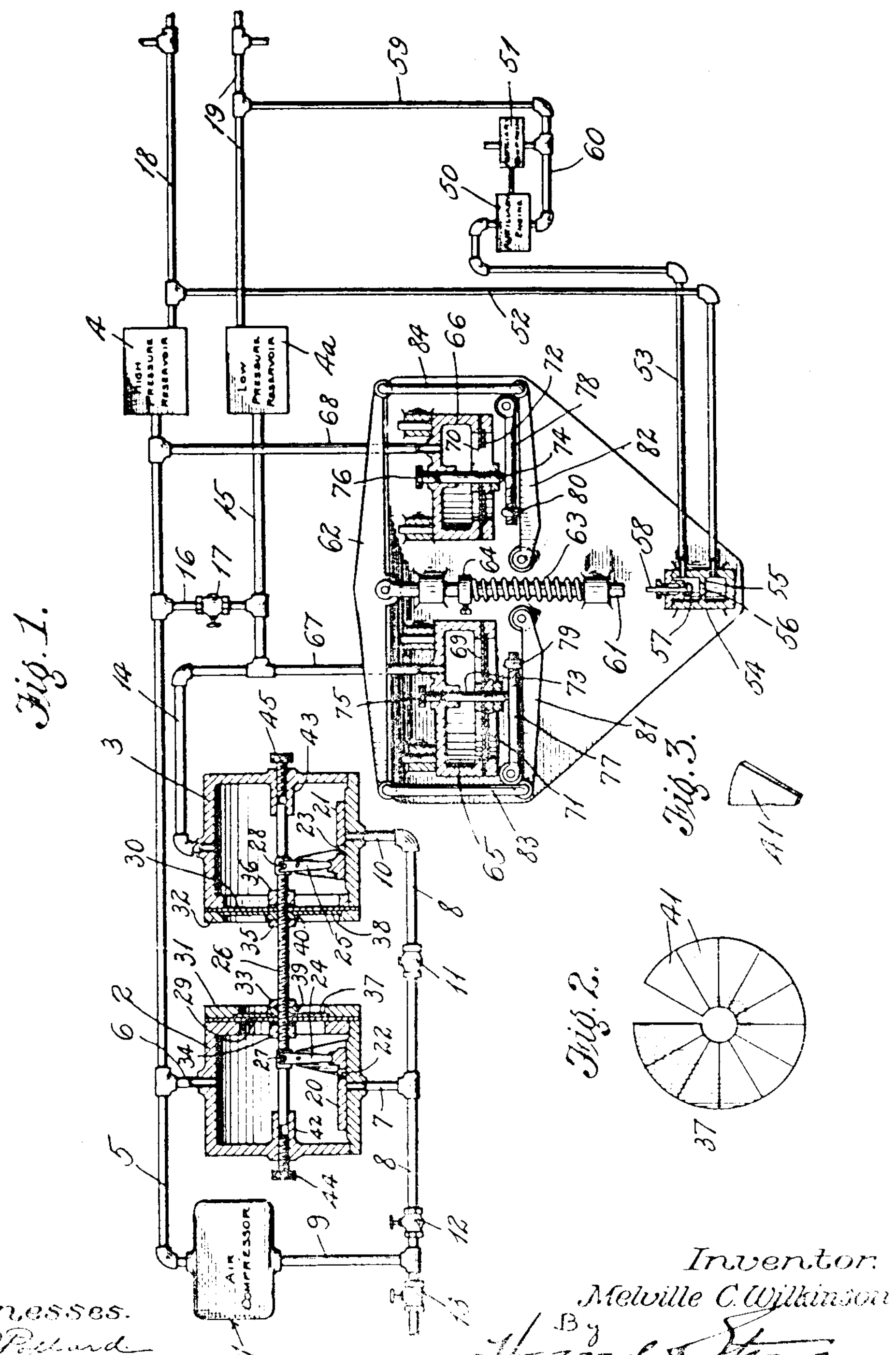
APPARATUS FOR COMPRESSING AND DISTRIBUTING AIR UNDER PRESSURE.

APPLICATION FILED FEB 13, 1911.

1,017,835. Patented Feb. 20, 1912.



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APPARATUS FOR COMPRESSING AND DISTRIBUTING AIR UNDER PRESSURE.

1,017,835. Specification of Letters Patent. Patented Feb. 20, 1912.

Application filed February 13, 1911. Serial No. 608,214.

To all whom it may concern:

Be it known that I, MELVILLE C. WILKINSON, a citizen of the United States, residing at Paloverde Valley, in the county of Riverside and State of California, have invented new and useful Improvements in Apparatus for Compressing and Distributing Air Under Pressure, of which the following is a specification.

This invention relates to improvements in apparatus for compressing, conveying and distributing compressed air and utilizing the same as a motive force to drive machinery at a more or less distant point from the source of power, the exhaust of the working machinery, still under pressure, being returned and recompressed for use. It has been customary in mechanisms of this character either to vary the speed of the compressor, to suit the load of the working machinery, or to allow the excess air compression over the amount used in the working machinery, to escape from the high pressure side to the low pressure side and become lost.

It is an object of this invention, among other things, to operate the compressor at a reasonably uniform speed, and to generate only the power required to operate the working machinery and further to produce an air compressing and utilizing mechanism which will require no cooling device.

It is also an object of the invention to reduce the size of the mechanism in proportion to the power generated and to utilize an unevenly distributed power to produce an even application of the power.

The invention is also designed to maintain a given ratio of pressure in the two sides of the system, the mechanism of this invention having high and low pressure sides provided with means to maintain the desired ratio between the pressure of the two sides.

It is a further object of the invention to provide means for automatically replacing any loss of air due to leakage in the mechanism, also to compress and utilize air in such a manner that the effects commonly due to heat or pressure generated will not vary the action of the machine.

In the drawing forming a part of this specification: Figure 1 is a diagrammatic plan view, partly in section, of the mechanism. Fig. 2 is an enlarged side view of one of the diaphragm protecting plates showing the sectional divisions. Fig. 3 is a perspective view of one section of said plate as shown in Fig. 2.

The invention and its operation will now be described in detail, reference being had to the parts illustrated in the drawings, in which 1 indicates an air compressor of any preferred design (construction not shown as not being a portion of the invention) operated by a prime mover.

2 and 3 are valve chambers, one of which as 2 is connected to the outlet side of the compressor by pipes 5 and 6, and to the inlet side of the compressor by pipes 7, 8 and 9, while chamber 3 is connected to the inlet side only of the compressor 1 by the pipes 10, 8 and 9. A high pressure reservoir 4 forms a portion of the high pressure side of the system and is connected to the air compressor 1 by the pipe 5.

A low pressure reservoir 4^a forms a portion of the low pressure side of the system and is connected to the chamber 3 by the pipes 14 and 15. The pipe 8 extends beyond its connections with both pipes 7 and 9; in it are mounted the valves 12 and 13 adjacent to its connection with pipe 9. A check valve 11 is also mounted therein intermediate its connection with pipes 7 and 10.

The high and low pressure sides of the system are connected by a pipe 16 intermediate the pipes 5 and 15, the passage therethrough being controlled by the valve 17. A pipe 18 extends from the high pressure reservoir 4 to the mechanism, not shown, which is to be driven or operated by the air compressed by the compressor 1. The pipe 19 extends from the driven mechanism to the reservoir 4^a, the exhaust air of the mechanism being returned therethrough to the low pressure side of the system.

The valve chambers 2 and 3 are provided with slide valves therein as 20 and 21, these slide valves are provided with ports 22 and 23 adapted to be brought opposite the ends of the pipes 7 and 10 when said pipes are to be opened. The valves 20 and 21 are operated respectively by levers 24 and 25 pivotally mounted at a suitable point intermediate their lengths, upon bearings in standards or supports within their respective chambers. The ends of the levers 24 and 25, opposite their connection with the valves 20 and 21 are preferably slotted, said slots engaging pins 27 and 28 secured to reciprocating rod or plunger 26, the ends of

which extend into the chambers 2 and 3 as shown in Fig. 1. The said rod or plunger 26 passes through the adjacent sides of the chambers 2 and 3 and the ends thereof project into, and are guided by, elongated bearings or guides 42 and 43 formed in the outer walls of the chambers 2 and 3. Set screws 44 and 45 projecting into the outer portions of said bearings make it possible to adjust and limit the throw of said rod 26. Portions of these sides of the chambers 2 and 3 through which the rod 26 passes, consist of diaphragms of leather, or other suitable material, concentric with the rod 26 and fastened to it by the nuts 33, 34, 35 and 36 which engage screw threads on the said rod 26; the outer edges of the diaphragms being secured to their respective casings by collars or clamping rings 31 and 32.

The diaphragms 29 and 30 are protected from bulging or distortion, under the action of the air within the chambers 2 and 3, by means of composite plates 37 and 38 which are mounted on the exterior sides of their respective diaphragms. The outer edges of the plates engage grooves in the inner edges of the collars or clamping rings 31 and 32, and the inner edges of the plates engage grooves in the outer edges of washers 39 and 40 which are situated just inside the nuts 33 and 35. These composite plates are built up of a multiplicity of sections, as 41, which are clearly shown in Figs. 2 and 3 of the drawings. The mounting of these segments 41 of each plate 37 and 38 is sufficiently loose in the grooves engaging their outer and inner edges to permit of their articulation under the action of the diaphragms 29 and 30 in moving the rod 26.

The active areas of the diaphragms 29 and 30 are in inverse ratio to the pressures which are to be maintained in the two sides of the system. I prefer to maintain a ratio of two (the pressure in the high pressure side of the system) to one (the pressure in the low pressure side of the system); or a pressure of 200 pounds absolute per square inch in the high pressure side, and 100 pounds absolute per square inch in the low pressure side.

In the operation of this mechanism, should there be no available supply of air already compressed with which to charge the two sides of the system to their working pressures, air is introduced in two stages. The first stage is to supply a difference of pressure in the two sides of the system sufficient to operate the mechanism for supplying loss by leakage which is hereinafter described, and the second stage is to vitalize the mechanism for supplying loss by leakage to compress atmospheric air into the system to bring the pressure in the two sides of the system up to the working pressure. The air compressor 1 is first caused to draw in air from the atmosphere by opening the valve 13 and closing the valve 12, this air is forced into the pipe 5 and the high pressure side of the system, a portion of it passing into the low presure side of the system through the pipe 16, the valve 17 being also open. Thus the pressure will increase equally in both sides of the system until a predetermined pressure is obtained, say 60 or 70 pounds per square inch, at which time the valves 13 and 17 should be closed and the valve 12 opened. This primary pressure against the diaphragm 30, in chamber 3, which has twice the area of diaphragm 29 in chamber 2, will cause the diaphragm 30 to force the rod 26 against the set-screw 44. This movement of the rod 26 will cause the lever 25 to move the valve 21 and bring the port or aperture 23 therein over the end of the pipe 10; this same movement of the rod 26 acting through the lever 24 will cause the valve 20 to assume a position to close the pipe 7. The continued action of the air compressor 1 will then draw air from the chamber 3 and the low pressure side of the system through the pipes 10 and 8, the check valve 11 and pipe 9 into its inlet side and deliver it into the pipe 5, chamber 2 and the high pressure side of the system. The pressure in the high pressure side of the system will increase and the pressure in the low pressure side will decrease until the increased pressure in the chamber 2 upon the smaller diaphragm 29 is able to exert a counter pressure upon the said diaphragm 29 to that received by the larger diaphragm 30 in the chamber 3.

When the pressure in the chamber 2 is able to equalize and a little more than offset the pressure upon the diaphragm 30, the rod 26 will be moved in the opposite direction toward the set screw 45, causing the valve 21 to move and cut off the pipe 10 from the chamber 3, and a similar movement of the valve 20 will cause it to register its port 22 over the end of the pipe 7. The air in the low pressure side of the system is thus cut off from the air compressor 1 and the air in the high pressure side of the system will be admitted to the inlet side of the air compressor 1 through the pipes 7 and 8 and 9. The check valve 11 prevents this high pressure from escaping into the low pressure side of the system. When the two sides of the air compressor 1 are thus connected together the compression of air ceases and the air idles through the compressor under the same pressure thus relieving it of work. When air has been transferred from the high pressure side of the system to the low pressure side so that the ratio of pressure in the two sides is disturbed, the diaphragms 29 and 30 will automatically adjust the valves 20 and 21 to regain the proper ratio of pressures.

In using compressed air machinery of this type, it becomes necessary from time to time, to introduce additional air to offset that lost by leakage, or from any other cause by which the pressure in the two sides of the system may fall below the desired working pressures. I accomplish this process, by the use of auxiliary compressing means consisting of an auxiliary air engine 50 operating an auxiliary air compressor 51. The auxiliary air engine 50 receives air from the high pressure side of the system through the pipes 52 and 53, a chamber 54 containing a valve mechanism being interposed in said piping. The chamber 54 has a web or wall 55 mounted therein, said web or wall having an aperture therethrough forming a port, the surrounding web or wall forming a valve seat 56. A valve 57 is adapted to close against said valve seat and prevent the passage of air therethrough. The pipe 52 enters the chamber below the valve; the pipe 53 enters the chamber above the valve. The valve has a stem 58 which passes outwardly through the wall of the casing to a suitable distance beyond the same. The valve stem 58 surrounded with a stuffing box to prevent escape of air from within the casing 54. When the valve 57 is raised from the valve seat 56 air will pass through the port in the web and actuate the auxiliary engine 50 and thereby the auxiliary air compressor 51, which will then take in air from the atmosphere and force it into the low pressure side of the system through the pipes 59 and 19. The exhaust of the auxiliary air compressor also passes to the low pressure side of the system through the pipes 60, 59, and 19. The said valve 57 is operated at proper times by a plunger or rod 61 mounted in alinement with the valve stem 58. Said plunger 61 is capable of being reciprocated by a floating lever 62 with which it is pivotally connected. The plunger 61 reciprocates in suitable bearings projecting from the framing of the mechanism, and a coiled spring 63 surrounding a portion of said plunger engages one of said bearings at one end and at the other end an adjustable collar 64 secured to the plunger 61. Said spring 63 tends to force the plunger 61 away from the valve stem 58 of the valve 57. The plunger 61 holds the valve 57 seated against the action of the spring 63 when there is ample pressure in the two sides of the system, through the action of diaphragms 69 and 70 in the chambers 65 and 66, which are mounted adjacent to the floating lever or yoke 62. The chambers 65 and 66 are connected by piping 67 and 68 with the low and high pressure sides of the system respectively. The diaphragms 69 and 70 in said chambers 65 and 66 have the same ratio in pressure receiving areas as exists between the diaphragms 30 and 29 in the chambers 3 and 2, so that the pressures in said chambers 65 and 66 will act equally upon the valve controlling plunger 61. The said diaphragms 69 and 70 are mounted in the walls of the chambers 65 and 66 in a manner similar to that of diaphragms 29 and 30 in the chambers 2 and 3, and they are protected by composite plates 71 and 72 similar to the plates 37 and 38 in the chambers 2 and 3. These diaphragms 69 and 70 each carry plungers 73 and 74 attached to their centers, in a manner similar to the attachment of the rod 26 to the diaphragms 29 and 30. Said plungers are controlled as to the extent of their inward movement by set screws 76 and 75 which act as stops. The exterior ends of said plungers engage levers 77 and 78 which are pivotally mounted at their outer ends upon the framing of the mechanism, while their inner ends are screw threaded and carry adjustable ring nuts 79 and 80, which bear upon levers 81 and 82, also pivotally mounted on the framing of the mechanism. The outer ends of said levers 81 and 82 are connected by links 83 and 84 with the ends of the floating lever or yoke 62. The pressure in the two sides of the system acting within the chambers 65 and 66 will actuate the diaphragms 69 and 70 and the plungers 73 and 74 carried by them, will operate the system of levers intermediate the plungers and floating lever 62, and the latter will force the plunger 61 in the proper direction for closing the valve 57 against its seat 56. Any less pressures than the desired working pressures in the two sides of the system will not be capable of compressing the spring 63 sufficiently to close the valve stopper against the seat, so that where the pressures in the two sides of the system are less than the working pressures desired, the spring 63 will expand and carry the rod 61 away from the stem 58 of the valve stopper 57, and air from the high pressure side of the system will first force the valve 57 away from its seat 56 and then pass said valve. As long as this said valve is open the air from the high pressure side of the system will pass through the pipe 52, casing 54 and pipe 53 to the auxiliary engine 50 and actuate said engine and auxiliary air compressor 51 and compress atmospheric air and force it into the low pressure side of the system. It will then be proportionately transferred to both sides of the system until the increased pressures acting on the two diaphragms 69 and 70, cause them to close the valve 57 upon its seat 56 and stop the auxiliary engine 50 and auxiliary air compressor 51.

Where the power supplied by this mechanism is applied to such purposes as the propelling of vehicles, at a certain speed the propelling motion will consume the full volume of the capacity of the air compressor 1, at the full difference of the pressure between

the high and low pressure sides of the system. An increased speed of the motors will take more than the capacity of the compressor at full pressure and the pressure on the high pressure side of the system will be reduced while that of the low pressure will increase; the mechanism set forth is admirably adapted to providing means that will supply loss by leakage at that time, no matter what may be the ratio of pressure on the two sides of the system, since the said mechanism involves the combination of the resultant actions of two pressures converted to a single function. Should the pressures exerted in the two chambers 65 and 66 be disturbed so that the resultant pressure communicated to rod 61, is not equal to the pressure when the ratio is maintained in the two sides of the system, this difference can be compensated for by the adjustment of the ring nuts 79 and 80 upon the levers 77 and 78. The adjustment of these same ring nuts will also compensate for the deterioration, through long use, in the spring 63.

It will be apparent that where the two sides of the system are filled with air to the desired pressures and the motors or other machinery which may be operated by the air, from the system are running, the rod or plunger 26 operating the valves in the chambers 2 and 3 will move backward and forward, alternately opening and closing the said valves to admit air to the compressor from either the high or the low pressure sides of the system as required, and when the motors or other machinery to be actuated are idle, the valve 20 will remain open and the valve 21 closed. When the motors are working and taking the full supply of air furnished by the compressor 1, the valve 20 will remain closed and the valve 21 open. Any intermediate use of the motors will cause the mechanism to move back and forth to maintain the given ratio of pressure in the two sides of the system, and the work that will be required of the compressor will depend upon the length of time the valve 21 is open and the valve 20 closed over the time when the said valve 20 is open and the valve 21 is closed.

It is believed that the operation of the mechanism will be fully understood from the above description since the operation of its various parts has been necessarily brought out in connection with said description.

What I claim is:—

1. An air compressing and distributing mechanism, comprising a compressing means, a system of piping connected with the induction and the eduction sides of said compressor, a movable member interposed between the sides of said system and operable by the relative pressure in the two sides of the system for causing the compressor to take air from either side of the system and thus maintain the pressure in the two sides at a given rate with respect to each other.

2. An air compressing and distributing mechanism, comprising an air compressing device, a system of piping and reservoirs connected with the compression side of said device, a system of piping and reservoirs connected with the suction side of the said device, a movable member associated with the system and adapted to be actuated by the pressures from both sides of the system whereby a given ratio between the two systems may be maintained.

3. An air compressing and distributing system, comprising an air compressor, one side of said system being connected with the eduction side of the compressor while the other side of the system is connected with the induction side of the compressor, a reciprocating member associated with the system and having means for receiving the different pressures from the two sides of the system, and means operated by the said reciprocating member for maintaining a given ratio of pressure between the two sides, in accordance with the work performed by the air upon the compression side of the said system.

4. An air compressing and distributing system, comprising high and low pressure reservoirs, an air compressor connected with each by a suitable piping, and capable of drawing air from either reservoir in accordance with the amount of work performed, a reciprocating member associated with the system, valves operated by the said reciprocating member, the said valves being interposed in said piping and arranged to be moved by the reciprocating member in accordance with the pressure exerted upon the same from the two sides of the system, and means coöperating with the high and low pressure sides of the system capable of introducing additional air to the system whenever leakage occurs.

5. A compressed air system, comprising high and low pressure sides, a compressor interposed between them, a valve for connecting the low pressure side with the compressor when the pressure therein is depressed below the usual rate maintained and a second valve connected with the high pressure side, a reciprocating member connected with both of said valves, means carried by the said reciprocating member adapted to receive the differing pressures from the two sides of the system whereby the member will be so moved as to control the valves for maintaining the desired ratio between the sides of the system.

6. An air compressing and delivery system comprising high and low pressure connections, an air compressor interposed be-

tween them, and arranged to compress into the high pressure side from the low pressure side, valves controlling the delivery of pressure from the low pressure side, and the maintaining of the same in the high pressure side, flexible diaphragms of different areas controlling the action of said valves in accordance with the ratio of pressures to be maintained in the different sides of the system, and means for introducing additional air to the system controlled by the pressures maintained in the two sides of the system.

7. An air compressing and delivering system comprising a compressor, piping and intermediate connections connected with the inlet and exhaust sides of the compressor forming a low pressure system and a high pressure system, equalizing valve mechanisms interposed in said systems, one of said valve mechanisms being connected with the low pressure side only, while the other is connected with both low and high pressure sides, a reciprocating member extending into both valve mechanisms and capable of operating the same simultaneously, pressure means having different pressure receiving areas affecting the said valve mechanisms, the pressure receiving areas in the low pressure side being greater than that in the high pressure side whereby a higher rate of pressure is maintained through the action of the said valve mechanisms upon the high pressure side.

8. An air compressing and distributing mechanism comprising high and low pressure sides, a compressor interposed between the two, an auxiliary compressing means connected with the low pressure side of the system, means for furnishing actuating air pressure thereto from the high pressure sides of the system, means controlled by the pressure in both sides of the system for admitting the air pressure to said auxiliary means.

9. An air compressing and distributing mechanism comprising a compressor having a system of piping connected with the outlet side of the compressor for holding the high pressure generated by the air compressor, a system of piping connected with the inlet side of the compressor for holding the low pressure maintained in the apparatus, equalizing valves connected with the low pressure system, one of which valves is also connected with the high pressure side of the system, an auxiliary compressing means connected with the low pressure side of the system, a valve mechanism for introducing air from the system to actuate said auxiliary compressor, and means actuated by the pressures from the high and low pressure sides of the system for controlling the said valve and permitting the operation of the auxiliary compressor when additional air is required in the system.

10. An air compressing and distributing system having a high pressure side and a low pressure side connected with the eduction and induction sides of the air compressor respectively, an air compressor, valve mechanisms controlling the ratio of pressure between the sides of the system, diaphragms of different areas for actuating said valves in accordance with pressure exerted upon them, the said valves being formed of flexible material secured to the valve casings, a reciprocating member operating the said valves and connected with the said diaphragms, and segment plates movably mounted outside the said diaphragms for protecting the latter against undue stretching and distortion.

11. An air compressing and distributing mechanism comprising an air compressor, systems of piping connected with the opposite sides thereof for holding air under high and low pressure, valve means for maintaining the proper ratio between the sides of the system, an auxiliary air compressing mechanism connected with the system, a valve mechanism for controlling the same, a plunger for operating said valve mechanism, levers for actuating the said plunger, and pressure receiving diaphragms connected with the high and low pressure sides of the system for operating said levers and the said plungers in accordance with the pressure in the system.

12. An air compressing and distributing mechanism comprising a main air compressor, high and low pressure systems of piping connected with the inlet and outlet ports of said compressor, valve chambers connected with the said systems of piping for maintaining a given ratio in the pressure upon each side of the compressor, an auxiliary compression mechanism for adding more air to the system when the pressure therein has fallen below normal condition, a valve controlling the same, a spring pressed plunger for operating said valve, a floating lever carrying the same, adjustable levers for operating said floating lever, yielding diaphragms mounted in pressure casings, pressure casings carrying the same, one of which is connected with the low pressure side of the system while the other is connected with the high pressure sides thereof, the said diaphragms having differing pressure areas corresponding inversely with the ratio maintained between the high and low pressure sides of the system, plungers carried by the said diaphragms engaging the said adjustable levers, and segment plates movably mounted outside the diaphragms for protecting them against undue strain and distortion.

13. An air compressing and distributing mechanism, comprising a compressor, a system of piping connected with the eduction

side thereof, a system of piping connected with the induction side thereof, valve mechanisms connected with each side of said system, one of said valve mechanisms being also connected with the other side of the system, a reciprocating member extending into both of said valve mechanisms for operating the said valves, and diaphragms of different areas connected with said reciprocating member for influencing the same in accordance with the ratio between the pressures in the two sides of the system.

14. An air compressing and distributing mechanism, comprising a compressor, systems of piping connected with the induction and eduction sides of said compressor, a valve mechanism having connections with both sides of the system, a second valve mechanism having connections with the induction side of the system only, a reciprocating rod extending into both of said valve mechanisms, valves in each mechanism adapted to be controlled by the said rod, flexible diaphragms connected with the said rods and having suitable areas exposed to the pressures of each side of the system, one of said diaphragms being larger than the other to maintain a higher pressure upon one side of the system than upon the other, protecting means carried by the said rod for protecting the said diaphragms, and means for adjusting the limit of travel of the said reciprocating rod, the whole structure being such that the pressures upon the two sides of the system will be automatically maintained at a given ratio with respect to each other.

In witness that I claim the foregoing I have hereunto subscribed my name this 4 day of February, 1911.

MELVILLE C. WILKINSON.

Witnesses:

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G. A. Parker.

Oct. 27, 1931. R. E. BRUCKNER 1,829,261

POWER TRANSMITTING MECHANISM

Filed Sept. 22, 1926 3 Sheets-Sheet 1

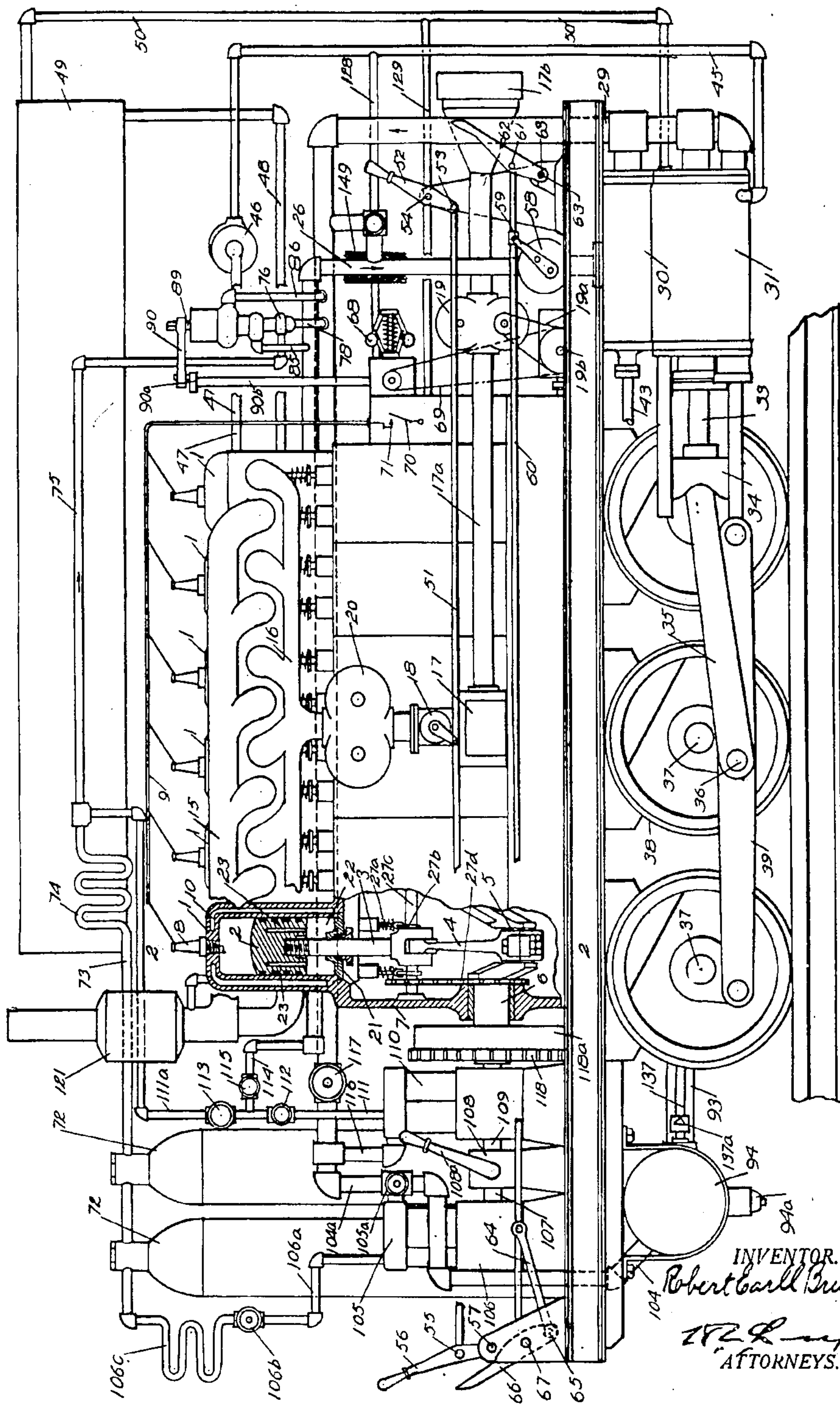

FIG. 1.

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POWER TRANSMITTING MECHANISM

Filed Sept. 22, 1926 3 Sheets-Sheet 2

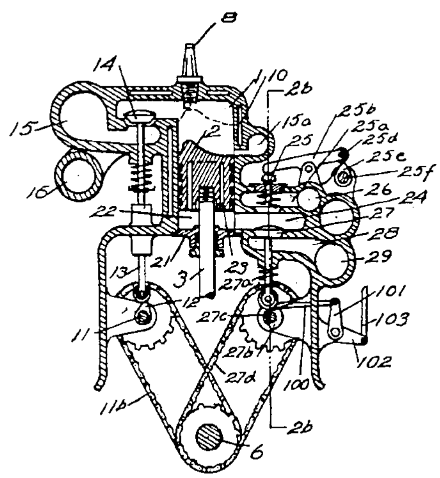


FIG. 2.

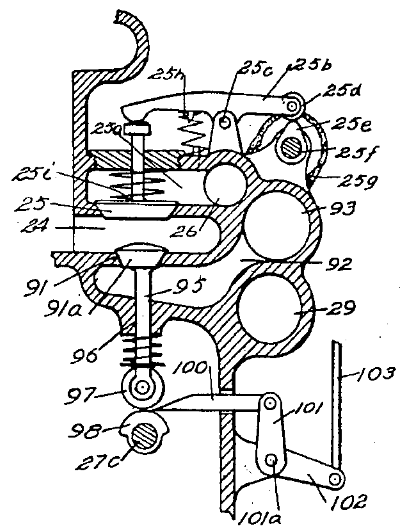


FIG. 2a.

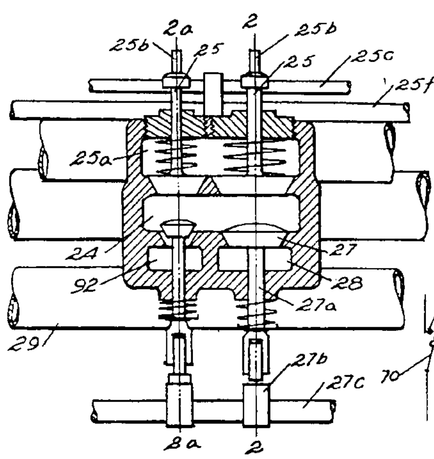


FIG. 2b.

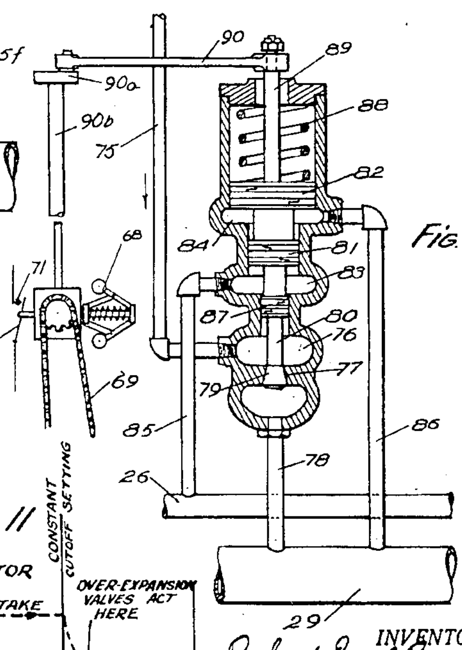


FIG. 3.

FIG. 10.

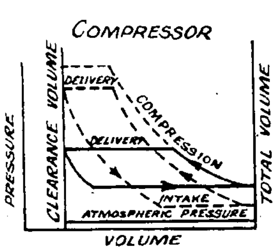
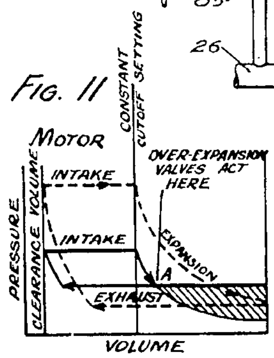


FIG. 11



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POWER TRANSMITTING MECHANISM

Filed Sept. 22, 1926 3 Sheets-Sheet 3

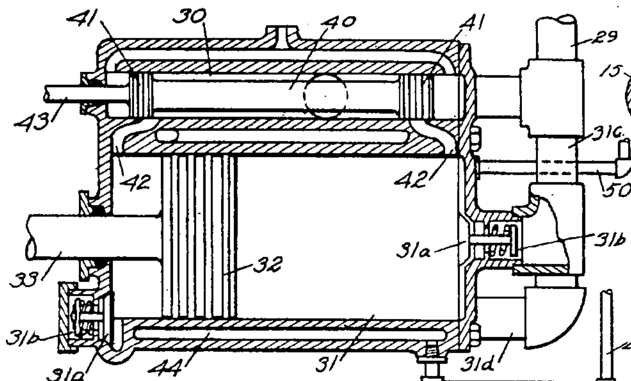

FIG. 4.

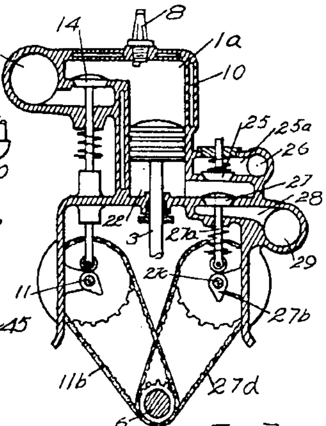

FIG. 7.

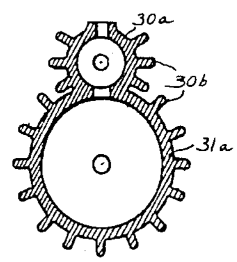

FIG. 8.

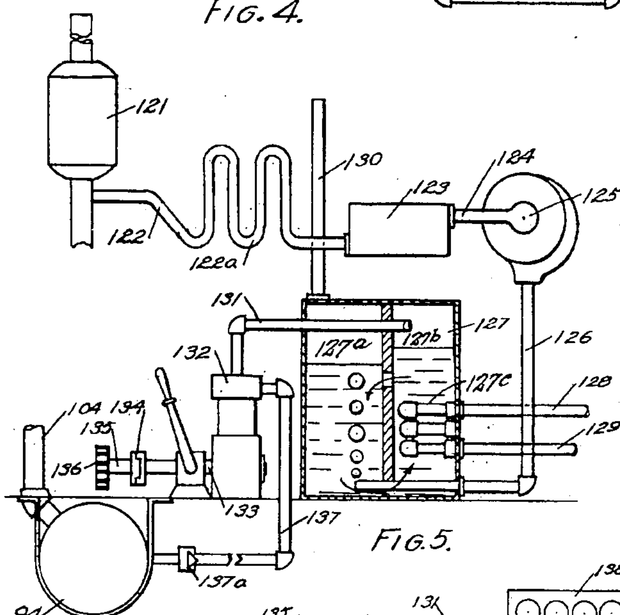

FIG. 5.

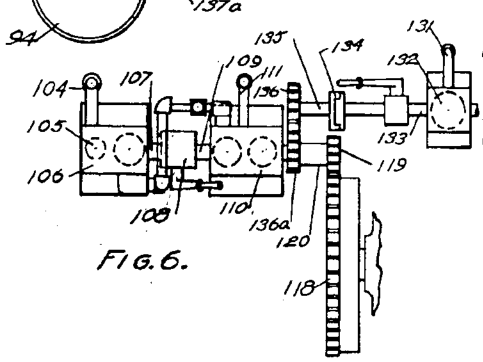

FIG. 6.

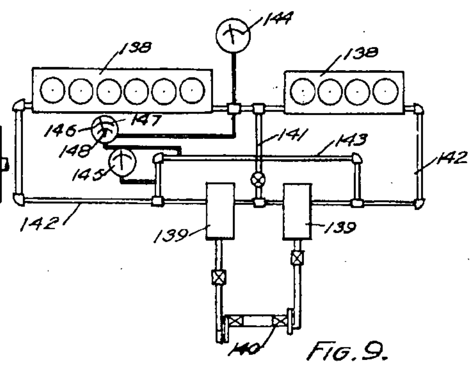

FIG. 9.

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Patented Oct. 27, 1931

1,829,261

UNITED STATES PATENT OFFICE

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POWER TRANSMITTING MECHANISM

Application filed September 22, 1926. Serial No. 137,158.

The present invention comprises a system for the pneumatic transmission of the power from a prime mover, such as a combustion engine, to a load requiring sensitive variation of speed and torque, and usually also quick reversal, such as rolling mills, locomotives and the like.

The objects of the present invention are: to reduce the size of compressor required to absorb the power of a given engine; to increase the compressor speed, eliminating the necessity for speed-reduction between the engine and compressor; to increase the efficiency of transmission, primarily by reducing the number of working parts between the expanding engine fluid and the transmission fluid being compressed; to reduce the danger of explosions of lubricant vapor; to reduce the bulk of the pneumatic motor required for complete expansion of the transmission fluid; to increase the efficiency at starting, with motor cutoff set late to provide sufficient starting torque; to eliminate the chilling of the motor lubricant due to expansion of the transmission fluid; to simplify the governing system, providing torque and speed adjustment by operator control of engine speed to vary the motor speed. Other objects of the present invention will be evident from the following description.

While described primarily as cooperating with an internal combustion engine of the Otto type, the present invention is applicable to combustion engines generally. The specific manner of use with any particular combustion engine will, from the following description, be evident to those skilled in the design of the engine.

Features and details of the invention will appear from the specification and claims.

A preferred embodiment of the invention is illustrated in the accompanying drawings as follows:

Fig. 1 shows a side elevation of a locomotive, partly in section, embodying the invention.

Fig. 2 a sectional view of one of the engine cylinders and its valve mechanism on the line 2—2 in Fig. 2*b*.

Fig. 2*a* a section on the line 2*a*—2*a* in Fig. 2*b*.

Fig. 2*b* a section on the line 2*b*—2*b* in Fig. 2.

Fig. 3 a sectional view of the constant fluid quantity controlling mechanism.

Fig. 4 a sectional view of the pneumatic motor.

Fig. 5 an elevation of a fluid replenishing apparatus.

Fig. 6 a plan view of the starting compressor, starting motor and replenishing compressor.

Fig. 7 a central section of an alternative form of engine.

Fig. 8 a central section of an alternative form of pneumatic motor cylinder.

Fig. 9 a diagrammatic view illustrative of the operation of the mechanism and an alternative application of the same.

Fig. 10 a compressor diagram.

Fig. 11 a motor diagram.

The primary motor is preferably made up of a plurality of cylinders 1. These are provided with pistons 2, the piston rods 3, connecting rods 4, and cranks 5 mounted on crank shaft 6. The crank shaft 6 is carried in bearings in a frame 7 on which the cylinders are mounted. Ignition plugs 8 are of the usual type and supplied through the wiring 9. A water jacket 10 of the usual construction is provided for the cylinder. An engine cam shaft 11 is driven by a sprocket gearing 11*b* from the crank shaft. The cam shaft is provided with cams 12 which actuate push rods 13, the push rods carrying inlet and exhaust valves 14, the exhaust valve appearing in the section in Fig. 2. The exhaust valves lead to an exhaust manifold 15 and an inlet manifold 16 is provided for the inlet. An auxiliary exhaust port 15*a* is provided for each of the cylinders, the cam timing as shown in Fig. 2 being arranged for a two-stroke cycle engine. A carburetor 17 is of the usual type and has its intake for air through a pipe 17*a* with the usual inlet opening 17*b* and a throttle valve 18 is provided at the discharge side of the carburetor in the usual manner. Preferably the engine is provided with super-

chargers 19 and 20, the supercharger 19 being driven through a chain gearing 19*a* from a shaft 19*b* driven from the engine shaft, the supercharger 20 being similarly driven from the engine cam shaft by a gearing (not shown).

The lower ends of the cylinders are closed by covers 21 providing the compressor chambers 22. The lower face of the piston is provided with annular fins 23 to increase the surface and consequently the heat transfer by way of the piston. The compressor chamber has a discharge port 24 and this leads to a passage 25*a* which is controlled by a discharge valve 25. The passage 25*a* leads to a discharge pipe 26. An inlet valve 27 controls the connection between the port 24 and an inlet passage 28 leading from an inlet pipe 29. The valve 27 is provided with a stem 27*a* which is operated by a cam 27*b* carried by a cam shaft 27*c* and the cam shaft is driven by a sprocket and chain 27*d* from the crank shaft.

Preferably also the discharge valve 25 is positively closed. This is accomplished by a rock arm 25*b* which operates on the upper end of the valve stem of the valve 25. The rock arm is mounted on a shaft 25*c* and is provided with a cam roller 25*d*. A cam 25*e* operates on the roller 25*d* to rock the shaft 25*c*. The cam is carried by a cam shaft 25*f* and the cam shaft is operated through a chain 25*g* from the cam shaft 27*c*. The closing cam 25*e* operates on the valve only momentarily and the valve is relieved of the weight of the rocker immediately through the action of the spring 25*h* so that immediately the valve is subjected to a pressure opening it for discharge it is relieved of the rocker arm pressure. The valve is preferably provided with a spring 25*i* which tends to lift the valve, the tension of the spring balancing the weight of the valve parts to make the valve sensitive.

The pneumatic motor may be of any preferred type. As shown the pipes 26 and 29 lead to and from a valve chest 30 of the motor, the valve chest being arranged on a cylinder 31. A piston 32 operates in the cylinder and is provided with a piston rod 33 extending to a cross head 34. A pitman 35 extends from the cross head to one of the cranks 36 carried on the driving axles 37. Driving wheels 38 are mounted on the axles and the different cranks are connected by the side rods 39. An ordinary slide valve, of the plug type, 40 is arranged in the valve chest and this has the controlling ends 41 operating over ports 42 leading to the cylinder. The valve is provided with a valve stem 43 which is connected with any suitable valve gear, preferably a reversing variable cut-off valve gear (not shown).

The cylinder preferably has a water jacket 44 and this water jacket has a pipe 45 leading from a centrifugal pump 46, the centrifugal pump being connected by a pipe 47 with the water jackets 10 of the engine cylinders. A pipe 48 leads to the water jackets 10 from a radiator 49 and a pipe 50 leads from the jacket 44 to the radiator so that the pneumatic motor cylinders are warmed by the heat from the cylinders of the internal combustion engine, this heat transfer being accomplished through the liquid kept in circulation by the centrifugal pump. The centrifugal pump may be driven by any convenient mechanism (not shown).

It is desirable that the throttle be operated from either end of the locomotive. To that end the throttle lever is connected by the rod 51 with the lever 52 at 53 below the pivotal mounting 54 of the lever 52 and at the opposite end of the locomotive at 55 with the lever 56. the lever 56 being mounted at 57 below the rod. In this way an inward movement of either the lever 52 or 56 in the same direction results in the same control movement of the throttle. It is desirable also for the purpose of quickly stopping the pneumatic motor to provide a throttle in the pipe 26 leading to the valve chest. Such a throttle 58 is here provided. It has a control lever 59 and here also it is desirable to control this from either end of the locomotive with a movement of a controlling lever (pedal actuated) in the same direction. A rod 60 is connected to the throttle lever, 59, and at 61 to a pedal lever 62, the pedal lever being pivotally mounted at 63 below point 61. At the opposite end of the rod 60 a link 64 connects the rod 60 with a pedal lever 66 at 65 below a pivotal mounting 67. Consequently a downward movement of the levers 62 and 66 results in a similar movement of the throttle lever 59.

It is desirable to supply the device with a safety governor for reasons hereinafter described. Such a governor is provided in the form of a centrifugal governing mechanism 68 which is driven by a chain 69 from the shaft 19*b*. The stem of the centrifugal governing mechanism operates upon a switch 70 acting on terminals 71 controlling the ignition system.

It will be understood that it is desirable to maintain a definite quantity of actuating fluid in the transmission system. This is accomplished in the present invention by a mean pressure controlling mechanism shown in detail in Fig. 3. Preferably the fluid medium is CO_2 and this is stored in liquid form in containers 72. A pipe 73 leads from the containers 72 to a warming coil 74, the coil 74 being warmed by atmosphere so as to convert the wet vapor given off from the containers into a comparatively dry gas. This coil is connected by a pipe 75 with a chamber 76. The chamber is connected through a valve opening 77 with a pipe 78.

the pipe 78 leading to the low pressure pipe 29. A valve 79 controls the passage 77. It is provided with a stem 80. The stem is connected to two operating pistons 81 and 82 subjected to pressures from chambers 83 and 84 respectively. The chamber 83 is connected by a pipe 85 with the high pressure pipe 26 and the chamber 84 by a pipe 86 with the low pressure pipe 29. The chamber 83 is closed against the chamber 76 by a closure plunger 87. A spring 88 exerts downward pressure on the valve and tends to open it while the pistons 81 and 82 tend to close it.

This apparatus is designed to maintain an approximately constant quantity of operating fluid in the circuit. The effective area of the pistons 81 and 82, therefore, should be so proportioned that the sum of the pressures on the pistons operating in conjunction with the spring affect this result. To this end the area of the piston 81 should be proportioned to the effective area of the piston 82 as the volume of the high pressure side is to the volume of the low pressure side, changing this proportion slightly to compensate for differences in absolute mean operating temperature of the fluid on the two sides. In this way the total quantity of fluid in the circuit may be maintained approximately constant. In order to make the device more sensitive I prefer to provide an extension 89 above the upper piston 82 and oscillate the pistons through an arm 90 actuated by a crank 90a mounted on a shaft 90b driven from the governor 68. In order to start the engine I prefer to utilize the compressor system as a motor for starting the engine. This can be readily accomplished by the valve mechanism illustrated in Figs. 2, 2a and 2b. An auxiliary starting passage 91 leads from the port 24 to a passage 92, the passage 92 leading by a pipe 93 to a low pressure receiver 94. The passage 91 is controlled by a valve 91a and this is provided with a stem 95 operating in a guide 96. The stem 95 has a cam roller 97 at its lower end which is adapted to be operated by a cam 98 on the shaft 27c. A wedge piece 100 is adapted to be interposed between the end of the roller 97 and the cam 98 so as to impart to the valve 91 action from the cam 98. The wedge piece is carried by a rock arm 101. The rock arm is mounted on a shaft 101a. A rock arm 102 extends from the shaft 101a and is operated by a control rod 103. When, therefore, it is desired to start the engine it is only necessary to throw in the wedge pieces 100. Some of these wedge pieces due to the timing will open at least one of the valves 91 thus exhausting fluid from the compressor chamber controlled by that particular valve, thus releasing the pressure under the piston of that particular unit. This will unbalance the units relatively to each other and the pressure fluid operating under the pistons of other units of the engine will force those pistons up, thus turning the engine over, the valves 91 being timed to operate the units in succession to accomplish this result in the ordinary manner, the opening of the valve 91 being coincident with the closing of the intake valve 27.

It will be observed that the constant-fluid controlling mechanism through the valve 79 will add fluid to the system to compensate for any fluid exhausted in the starting operation. In order that the receiver 94 may be certainly maintained at a pressure low enough to accomplish the starting should this be retarded, a blow-off valve 94a is provided operating at a desired maximum pressure in the receiver 94.

It is desirable to return the fluid collected in the receiver 94 to the storage containers, or bottles 72. To this end the following mechanism is provided: A pipe 104 leads from the receiver 94 to a valve chamber 105 of a compressor 106. The compressor 106 is driven from a shaft 107. The shaft 107 is driven from a shaft 109 through a clutch 108. The shaft 109 is a part of, or connected with, the power shaft of the starting motor 110. The motor 110 is fluid driven and gets its supply by way of a pipe 111. The pipe 111 is connected by a pipe 114 with the high pressure pipe 26. The pipe 111 is also connected by way of a pipe 111a with the pipe 75. A valve 112 is arranged in the pipe 111 and a valve 115 in the pipe 114 and a valve 113 in the pipe 111a, the connection of the pipe 114 with the pipe 111 being between the valves 112 and 113. When the engine is running and the starting motor is operated the valve 113 is closed and the valves 112 and 115 are open. Under these conditions, the exhaust from the starting motor 110 is carried by way of a pipe 116 to the low pressure pipe 29. A valve 117 is arranged in this pipe and is open as the motor is operated under the conditions just expressed. As the motor is operated and the compressor 106 actuated fluid is pumped from the receiver 94 and is discharged from the compressor through a pipe 106a to the containers, or bottles 72. The pipe 106a is provided with a valve 106b which may be closed when the compressor is idle and the pipe is also provided with a cooling coil 106c. It may be desirable to use the starting motor 110 to start the engine as distinguished from the valve action illustrated in Figs. 2, 2a and 2b. Under these conditions the valve 115 is closed and the valve 113 opened. Thus fluid direct from the pipe 75 is delivered to the starting motor 110 and under these conditions exhaust from the motor is delivered to the low pressure receiver 94 and the valve 117 is closed. A pipe 104a leads from the pipe 104 to the exhaust pipe 116 and a three-way valve 105a is arranged between the pipe 104

and 104a. The valve 105a is turned to close the passage into the valve chest 105 and open the connection between the pipe 104 and 104a so that the exhaust from the motor 110 takes place through the pipe 116, pipe 104a, pipe 104 to the low pressure receiver. A gear 118 is mounted on a fly wheel 118a of the crank shaft 6. A gear 119 driven from the shaft 109 of the engine operates on the gear 118. The gear 119 is controlled by any desired starting connection 120 which releases the connection with a driving speed of the gear 118.

It is desirable to replenish the system with the operating fluid so as to compensate for any leakage through the power motor, or compressors. This can be conveniently done when CO_2 is used as the transmission fluid by collecting the CO_2 from the exhaust of the engine. An apparatus designed for this purpose is as follows: The engine is provided with a muffler 121. Exhaust gases are drawn off from the exhaust pipe 15, this being between the engine and the muffler so as to get any advantage of back pressure from the muffler. A pipe 122 leads through a cooling coil 122a to a filter 123 and a pipe 124 leads from the filter to the intake of a fan 125. A pipe 126 leads from the fan to a separator 127. The separator is connected by pipes 128 and 129 with the water pipes 45 and 50 respectively so that the separator is heated sufficiently to facilitate its operation. The separator may be of any well-known design, either physical or chemical. A suitable device is shown in Fig. 5 in which the separator 127 is divided into two chambers 127a and 127b and designed to circulate a liquid through said chambers as indicated by the arrows, heat being delivered through the coil 127c, the coil being connected with the pipes 128 and 129. A solution of a mixture of sodium carbonate and sodium bicarbonate is used, and as heated by the coil 127c gives off some of its contained carbon dioxide, forming some sodium carbonate from some of the sodium bicarbonate. This action is facilitated by the suction from the compressor intake pipe 131 leading from the chamber 127b. Bubbles of exhaust gases are continuously issuing from the end of the pipe 126 into the chamber 127a. Some of the carbon dioxide in this stream of exhaust gases is absorbed in the solution, forming sodium bicarbonate, the remainder being vented with the inert exhaust gases through the pipe 130, the sodium bicarbonate formed giving up, as before stated, its carbon dioxide in the chamber 127b, and which is taken off through the pipe 131 to a compressor 132. The compressor is driven by a shaft 133 from a shaft 135 through a clutch 134. The shaft 135 is driven by a gear 136 from a gear 136a on the starting motor shaft. The CO_2 is delivered from the compressor by way of a pipe 137 to the low pressure receiver 94, a check valve 137a being provided to prevent a return flow from the receiver. It will be understood that the fluid accumulated in the low pressure receiver in this manner is returned to the containers, or bottles 72 by the compressor 106 in the manner heretofore described.

Operation

It will be noted in this transmission system the engine effort is transmitted pneumatically directly to the driving motor without the interposition of receivers so that there is an immediate response of the driving motor to changes in effort of the engine. Consequently in the normal operation of the locomotive the engine throttle 18 affords the only necessary control. This will be understood if the engine and compressor set be regarded as a pump pumping fluid from a low reservoir to a high reservoir and the driving motor set as a motor discharging fluid from the high to the low reservoir with a constant quantity of fluid in circuit. It is evident that for a given engine and compressor speed the pressure difference, and hence the torque on the driving motor, quickly and automatically increases as the motor speed decreases. On the other hand the driving motor speed will increase with a given engine speed where the load conditions of the driving motor decrease. But in any case the full range of speed may be accomplished by a variation of speed of the engine which under similar load conditions will be directly reflected by the change of the driving motor speed. With this system, it will be noted also, that by reason of the high back-pressure and the great variation in pressure difference between the two sides of the system it is possible to use a constant medium late cut-off while obtaining ample operating flexibility with practically a minimum expansion card in the motor for starting yet with more and more complete expansion as the motor speed increases and the pressure difference drops. If, however, a constant cut-off is used, means should be preferably provided for avoiding over-expansion and this would also occur with faulty operation where a variable cut-off is used.

I have, therefore, provided in the present device means for preventing over-expansion. This is illustrated in Fig. 4. Valves 31a lead from the ends of the cylinders into chambers 31b. These valves are of the check valve variety permitting flow into the cylinders but preventing flow outwardly. The chambers 31b are connected by pipes 31c and 31d with the low pressure side 29 of the transmission circuit. If, therefore, there is incipient over-expansion, fluid is immediately supplied to these valves from the low pressure side preventing such over-expansion and consequent work losses. While I have shown ordinary check valves, it will be understood that

any sensitive valve of the compressor type may be used.

These characteristics are illustrated by Fig. 10, which shows pressure-volume diagrams for the compressor, and by Fig. 11, which shows pressure-volume diagrams for the motor. The full-line pressure volume diagrams show the condition in which the motor is operating at low torque and high speed. The dotted diagrams show the condition in which the motor is operating at high torque and low speed, as when starting. The action of the over-expansion valves 31*a* is shown at A in Fig. 11, the shaded area showing the work otherwise lost.

It will be noted that the net work of the motor per revolution varies several hundred per cent, while the compressor work per revolution remains nearly constant at the optimum value for the given engine.

It will be understood that a variable cut-off is desired to get the maximum economy for differences in speed and torque. In the present system the total quantity of fluid remains constant in the transmission circuit. The maximum of economy is obtained by positioning the cut-off with relation to the working pressure difference for complete expansion. This point of cut-off can be readily established in the present system due to the fact that the mean pressure is constant by setting it with relation to this pressure difference.

In Fig. 9 I have shown diagrammatically a plurality of engines 138—138 delivering to driving motors 139 connected to a common load through a shaft 140. Fluid is conveyed from the engines through a connection 141 and returned through pipes 142, these pipes 141 and 142 corresponding to the pipes 26 and 29 of Fig. 1. The low pressure pipes 142 are coupled by a pipe 143. A gauge 144 indicates the high pressure and a gauge 145 the low pressure and a gauge 146 indicates the pressure difference. The differential gauge 146 is provided with two scales, 147 indicating the pressure difference and a scale 148 which designates cut-off settings for each pressure difference. Any variable cut-off valve mechanism commonly employed on locomotives, such as the Stephenson link may be used in connection with the motor, being connected as shown with the valve rod 43 (see Fig. 1). All that is necessary to obtain the proper cut-off setting for maximum economy is for the operator to set the cut-off to correspond with the pressure difference. It will be noted that the present system has a flexibility such that a plurality of engines of different sizes and speeds may be used operating upon the same driving motors substantially without change in the system.

It will be noted also that the direct connection between the compressor and the driving motor permits of the ready communication of the compressor heat to the driving motor thus preventing excessive low temperatures in the driving motor with the attendant lubricating and other troubles incident to such low temperature. This, in the present construction, is supplemented in the preferable form by the hot water circulating pipes extending to the jacket of the engine. This maintaining of the transmission fluid at a temperature preventing lubricating troubles both in the compressor and the driving motor is also supplemented by utilizing the engine piston as the piston for the compressor so that there is a direct heat transference from the engine to the compression side of the engine-and-compressor cylinder.

I prefer also to further lift the mean temperature of the transmission fluid in the closed circuit somewhat above atmospheric temperature by insulating the hot, or high pressure side while exposing the low pressure side to atmosphere or other heating medium. Such insulation is indicated at 149 (see Fig. 1).

By using expansible fluid under substantial pressure throughout the circuit, both in the high and low pressure sides it is possible to reduce the size of the compressor and still absorb the energy of the engine. This is because the volume ratio is comparatively small with a large pressure difference and this permits of a single-stage adiabatic compression with a large pressure difference. This makes possible the use of a compressor in which one end of the engine cylinder may be used for the compressor cylinder. This makes practical also a much higher compressor speed which is necessary to the practical operation of the engine. This high speed is not only facilitated by the nature of the compressor but by reason of the fact that the small volume ratio of compression gives a much longer open period to the outlet valves at the end of the compression stroke and also makes permissible a greater pressure difference at the two sides of the valves without objectionable loss. It is feasible, therefore, with this system to have a very much greater pressure drop between the high pressure and low pressure sides of the circuit with a comparatively simple single-stage mechanism.

While I have shown the compressor as formed directly in the engine cylinder and many of the advantages of my invention are dependent on such an arrangement it will be understood that in the broader aspects of the invention the compressor cylinder may be separated from the engine cylinder but preferably operated by the same reciprocating unit under the expansion of the engine fluid compressing the transmission fluid.

In order that the compressor may absorb the energy of the engine the quantity of transmission fluid should be maintained above a pre-determined minimum. The

quantity controlling device which automatically supplies to the transmission circuit any losses of fluid assures the complete absorption of the engine energy at all times by the compressor. It will be noted that the compressor also has a comparatively large clearance space. This clearance space should be such as to permit of the operation of the engine with no delivery from the compressor. Thus it is possible in instantly stop the motor in an emergency through the throttle 58 and to permit the engine to continue running. This is of particular importance also in the present system in that the engine will continue to operate with a full energy output with wide pressure differences so that the torque on the driving motor may have wide variations dependent on the load upon the driving motor with a constant full load torque on the engine. It will be understood, of course, that when the pressure difference in the system reaches a point that there is no fluid delivery the engine will then operate at the idling torque and it is for this reason largely that the safety governor is provided so that if this condition arises the ignition may be cut off and the running away of the engine prevented. It will be observed that all of these features which tend to automatically accommodate conditions of service assist in reducing the necessity for operative skill.

CO_2 forms a desirable medium for the transmission system because it is non-explosive, non-poisonous, cheaply obtainable, and also non-corrosive even in the presence of air. It has a boiling point much below atmosphere and will not freeze in the piping. It has the great advantage that its critical temperature is above atmosphere so that it can be liquefied by pressure alone while its critical pressure is low so that it can be carried in bottles in liquid form at less than one thousand pound pressure. By reason of its critical temperature and pressure it can be readily used for starting because atmospheric heat readily vaporizes it. Where a combustion engine is used the waste gases from the engine itself form a ready source of supply for the liquid.

It will be observed that by merely reversing the valve mechanism of the driving motor I may have regenerative braking. The motor then acts as a compressor drawing fluid from the low pressure side of the transmission circuit through the exhaust port of the main valve and through the relief valves 31*a* and compressing this fluid into the high pressure side of the circuit through the inlet port of the main valve. This regenerative braking energy of compression should preferably be utilized. Where this is desired, this is accomplished by driving the starting motor 110 from the transmission circuit. This has already been explained as being accomplished by connecting the high pressure side 26 through the pipe 114, valve 115, valve 112, pipe 111 with the motor 110 and exhausting the motor through the pipe 116 to the pipe 29, the energy of the motor being used to pump fluid from the low pressure receiver 94 to the bottles 72 as heretofore described, or in operating the compressor 132 in delivering replenishing fluid to the low pressure receiver. It will be observed that during the regenerative braking no change is made in the valve controls of the starting motor and the auxiliary compressors over their setting for operation directly from the systems where the engine supplies the power so that in this respect the apparatus operates automatically.

In Fig. 7 I have shown the apparatus as adapted to a four cycle power engine. In this device the only difference in structure is that the exhaust 15*a* is omitted from its cylinder 1*a* the cam shafts are timed for revolution with every other rotation of the crank shaft as in ordinary four cycle practice and the inlet valve 27 remains closed during the idle stroke and thus in effect unloads the compressor for this stroke.

It may be desirable also instead of using the water heating system for the driving motor cylinder to provide for heating it with atmospheric air. Under such condition the chest and cylinder 30*a* and 31*a* are provided with ribs 30*b* facilitating heat transfer from the atmosphere to the walls of the chest and cylinder.

It will be noted that the clearance of the compressor should be such that a constant compressor torque will continue through quite a wide range of variation of pressure difference and, therefore, of motor torque due to the expansion of the clearance volume and a diminished delivery with increasing pressure differences and this constant compressor torque should be slightly below the available maximum engine torque as has been before described.

It will be noted that in the present instance the compression stroke of the piston with relation to the combustion chamber is accomplished through the direct pressure on the lower face of the piston by the low pressure transmission gases. Thus there is a distinct power saving in as much as the mechanical connections are relieved of the strain in affecting the compression of the combustion gases.

What I claim as new is:—

1. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor driven by the engine, the compressor having a clearance without delivery creating only such back pressure as is overcome by the available torque of the engine; a motor; a normally closed pressure circuit between the compressor and motor;

means for maintaining the pressure in said circuit above atmospheric pressure; and means controlling the speed of the engine to control the speed of the motor.

2. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor driven by the engine; a motor; a normally closed pressure circuit between the compressor and motor, said compressor having a clearance creating a constant compressor torque throughout a range of variation of motor torque and within the maximum torque of the engine; means for maintaining said pressure above atmospheric pressure; and means controlling the speed of the engine to control the speed of the motor.

3. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor driven by the engine, the compressor having a clearance without delivery creating only such back pressure as is overcome by the available torque of the engine; a motor; a normally closed pressure circuit between the compressor and motor; means maintaining the pressure in said circuit above atmospheric pressure; means controlling the speed of the engine to control the speed of the motor; and a speed sensitive device stopping the engine at a predetermined speed.

4. In a power transmitting apparatus. the combination of an internal combustion engine comprising a reciprocating piston; a compressor comprising a reciprocating piston directly connected to the engine piston, a motor; a closed pressure gaseous fluid transmission circuit between the compressor and the motor; means maintaining the pressure in said circuit above atmospheric pressure.

5. In a power transmitting apparatus, the combination of an internal combustion engine comprising a cylinder and reciprocating piston; a compressor comprising a piston formed integrally with the engine piston and operating in the engine cylinder; and a closed pressure gaseous fluid transmission circuit between the compressor and the motor; means maintaining the pressure in said circuit above atmospheric pressure.

6. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor driven by the engine; a motor; a normally closed pressure circuit between the compressor and motor; means maintaining the pressure in said circuit above atmospheric pressure; means braking the motor by the fluid forced reversely through the circuit by the compressing action of the motor when the motor is subject to driving force and means utilizing the fluid reversely driven for starting the engine.

7. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor driven by the engine; a motor; a normally closed pressure circuit between the compressor and motor; means maintaining the pressure in said circuit above atmospheric pressure; means braking the motor by reversing the fluid in the circuit through the driving action of the motor; a starting motor; means circulating the fluid driven by the motor through the starting motor; and a compressor driven by the starting motor for starting the engine.

8. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor driven by the engine; a motor; a normally closed pressure circuit between the compressor and motor; means maintaining the pressure in said circuit above atmospheric pressure; and means controlling the speed of the engine to control the speed of the motor; a liquid fluid supply containing means connected to the circuit; starting means receiving fluid from the liquid supply; and means for gasifying said starting fluid before expansion in the starting means.

9. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor driven by the engine; a motor; a normally closed pressure circuit between the compressor and the motor; means maintaining the pressure in said circuit above atmospheric pressure; CO_2 in the circuit; means separating and collecting the CO_2 from the waste gases; and devices delivering the CO_2 collected to the circuit comprising a compressor driven by the fluid from the circuit.

10. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor driven by the engine; a motor; a normally closed pressure circuit between the compressor and the motor; means maintaining the pressure in said circuit above atmospheric pressure; CO_2 in the circuit; means separating and collecting the CO_2 from the waste gases; devices delivering the CO_2 collected to the circuit comprising a compressor driven by the fluid from the circuit condensing the collected CO_2 to liquid form; and a liquid container for storing the CO_2 in liquid form.

11. In a power transmitting apparatus, the combination of a variable speed internal combustion engine; a compressor driven by the engine; a motor normally controlled by the control of the internal combustion engine; a normally closed pressure circuit between the compressor and motor; means maintaining the pressure in said circuit above atmospheric pressure; and means maintaining a constant quantity of transmission fluid in the circuit.

12. In a power transmitting apparatus, the combination of a variable speed internal combustion engine; a compressor driven by the engine; a motor normally controlled by the control of the internal combustion engine; a normally closed pressure circuit be-

tween the compressor and motor; means maintaining the pressure in said circuit above atmospheric pressure; and means maintaining a constant quantity of transmission fluid in the circuit comprising a means of supply connected to the circuit, a valve controlling said means, and pressure devices connected into both sides of the circuit, and controlling said valve.

13. In a power transmitting apparatus, the combination of a variable speed internal combustion engine; a compressor driven by the engine; a motor normally controlled by the control of the internal combustion engine; a normally closed pressure circuit between the compressor and motor; means maintaining the pressure in said circuit above atmospheric pressure; and means maintaining a constant quantity of transmission fluid in the circuit comprising a means of supply connected to the circuit, a valve controlling said means, and pressure devices connected into both sides of the circuit, said pressure devices being proportioned in relation to the interior volumes of the respective sides of the circuit and the sum of the total forces on said pressure devices acting to control the valve.

14. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor driven by the engine; a motor; a normally closed pressure circuit between the compressor and motor; means maintaining the pressure in said circuit above atmospheric pressure; and automatic inlet valves opening from the low pressure side of the circuit to the motor to prevent over-expansion.

15. In a power transmitting apparatus, the combination of a variable speed internal combustion engine; a compressor driven by the engine; a motor normally controlled by the control of the internal combustion engine; a normally closed pressure circuit between the compressor and the motor; means maintaining the pressure in said circuit above atmospheric pressure; and means for maintaining the pressure of the transmission fluid at a mean value through the normal range of motor torque and speed sufficient to substantially balance the engine and compressor work per stroke.

16. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor driven by the engine; a motor; a normally closed pressure circuit between the compressor and motor; means maintaining the pressure in said circuit above atmospheric pressure; means maintaining a constant quantity of fluid in said circuit; and a differential gauge indicating pressure difference between the high and low pressure sides of the circuit, said gauge being provided with a scale indicating a desirable cut-off for each pressure difference.

17. In a power transmitting apparatus, the combination of a combustion engine; a compressor driven by the engine; a gaseous fluid actuated motor; and a closed circuit between the compressor and the motor; means maintaining a pressure above atmospheric on the low pressure side of the circuit, said engine and compressor being directly connected and the intake stroke of the compressor being directly opposed to a compression stroke of the engine whereby the pressure of the low pressure side of the circuit operates to compress the combustion fluid.

18. In a power transmitting apparatus, the combination of a combustion engine; a compressor driven by the engine; a fluid actuated motor; a closed circuit between the compressor and the motor; and means responsive to pressure changes in the low pressure side of said circuit automatically maintaining a pressure above atmospheric on the low pressure side of the circuit, said engine and compressor being directly connected and the intake stroke of the compressor being directly opposed to a compression stroke of the engine whereby the pressure of the low pressure side of the circuit operates directly to compress the combustible fluid.

19. In a power transmitting apparatus, the combination of a combustion engine comprising a piston; a compressor comprising a piston integral with the engine piston and driven by the engine; a fluid actuated motor; a closed circuit between the compressor and the motor; and means responsive to pressure changes in the low pressure side of said circuit automatically maintaining a pressure above atmospheric on the low pressure side of the circuit, the intake stroke of the compressor being directly opposed to the compression stroke of the engine whereby the pressure of the low pressure side of the circuit operates directly to compress the engine fluid.

20. In a power transmitting apparatus, the combination of a combustion engine comprising a piston; a compressor comprising a piston directly coupled to and reciprocating with the engine piston and driven by the engine; a fluid actuated motor; a closed circuit between the compressor and the motor; and means responsive to pressure changes in the low pressure side of said circuit automatically maintaining a pressure above atmospheric on the low pressure side of the circuit, the intake stroke of the compressor being directly opposed to the compression stroke of the engine whereby the pressure of the low pressure side of the circuit operates directly to compress the engine fluid.

21. In a power transmitting apparatus, the combination of a combustion engine comprising a piston; a compressor driven by said engine and comprising a piston reciprocat-

ing as a common unit with said engine piston, said compressor having a clearance volume sufficient to permit idling operation of the engine-compressor with all delivery from the compressor stopped; a fluid-actuated motor; and a pressure conduit between said compressor and said motor, the intake stroke of the compressor being directly opposed to the compression stroke of the engine, whereby the fluid trapped in the clearance volume of said compressor performs a substantial part of the work of compressing the combustion fluid, directly by pushing on the common piston assembly.

22. In a power transmitting apparatus, the combination of an internal combustion engine; a compressor, said engine and compressor having a common integral piston with the same working area for the engine and compressor operating in a common cylinder: a motor; a normally closed pressure circuit between the compressor and the motor; means maintaining the pressure in said circuit above atmospheric pressure; and a non-combustible fluid in the circuit, said piston being actuated to compress the combustible fluid by the intake of fluid entering the compressor.

In testimony whereof I have hereunto set my hand.

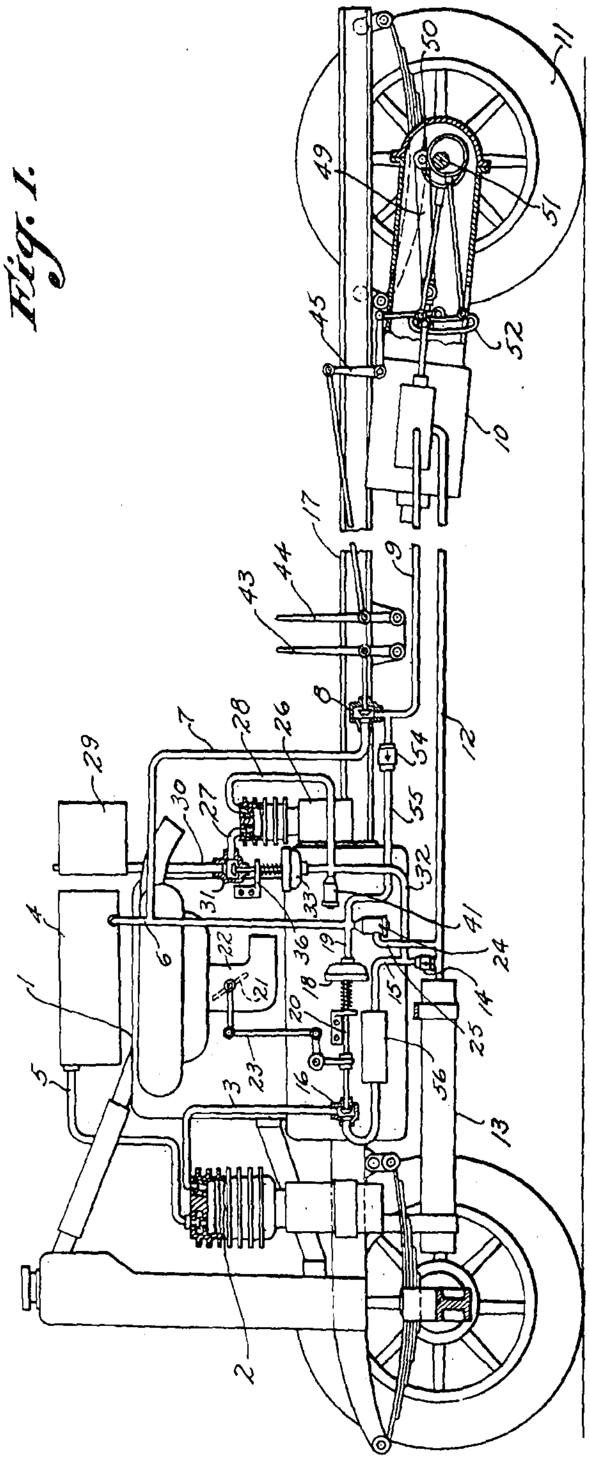
ROBERT EARLL BRUCKNER.

July 28, 1936. R. M. OTIS 2,049,078

FLUID TRANSMISSION

Filed July 9, 1934 2 Sheets-Sheet 1

Fig. 1.



INVENTOR:

Russell M. Otis

July 28, 1936.　　R. M. OTIS　　2,049,078

FLUID TRANSMISSION

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Fig. 3.

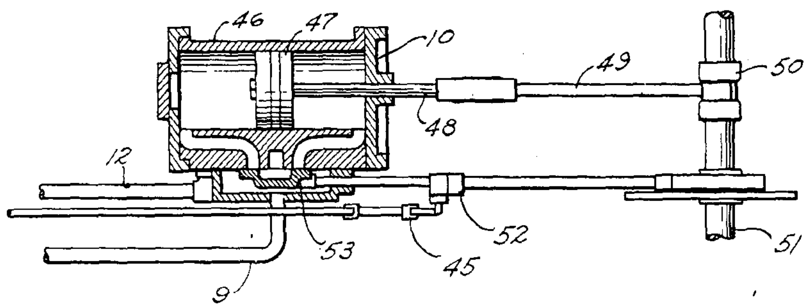
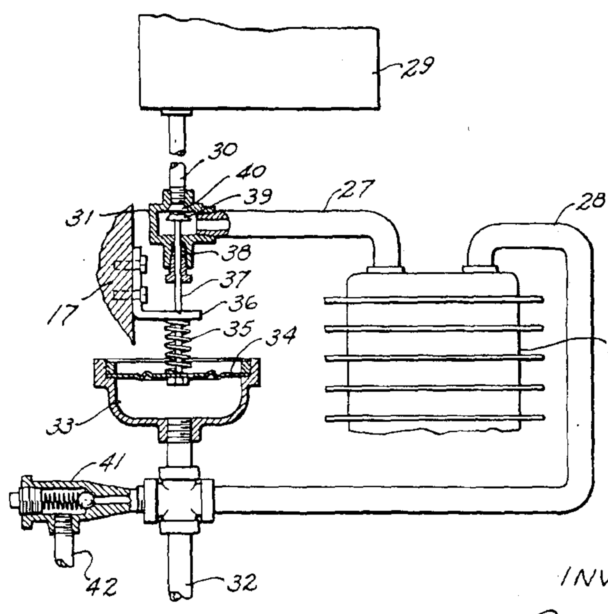


Fig. 2.



INVENTOR:

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Patented July 28, 1936 2,049,078

UNITED STATES PATENT OFFICE

2,049,078

FLUID TRANSMISSION

Russell M. Otis, Pasadena, Calif., assignor to Burt-Warner Power Corporation, Pasadena, Calif.

Application July 9, 1934, Serial No. 734,329

5 Claims. (Cl. 60—62)

My invention relates to fluid transmissions and particularly to closed circuit fluid transmission systems in which provision is made for braking of the driven member and in which a make-up means is provided to inject new fluid into the system to compensate for leakage.

An object of my invention is to make possible braking by the driving motor with consequent exhaustion of the main capacity of the low pressure side of the system without making it necessary for the make-up device to operate to inject new fluid into the system during this temporary depletion. Another object is to provide an arrangement of ports whereby the make-up device will come into use when there is a legitimate need for more fluid to be injected into the system. Other objects will appear from the disclosures in the specifications and drawings.

Fig. 1 is a drawing showing the fluid transmission as applied in an automobile.

Fig. 2 is a detail of the fluid make-up apparatus.

Fig. 3 is a detail of the fluid motor.

In the application to an automobile, illustrated in Fig. 1, an internal combustion engine 1 mounted on frame 17 drives compressor 2 also mounted on frame 17 which draws fluid through intake pipe 3 and its associated compressor check valve and compresses it into high pressure reservoir 4 through pipe 5 and its associated compressor check valve. The high pressure fluid passes through pipes 6 and 7, throttle 8, and pipe 9 to the valve-controlled fluid motor 10, which drives the wheels 11. The fluid is exhausted from the motor 10 through pipe 12, to which is connected low pressure reservoir 13. When the compressor is compressing, it takes air out of pipe 12 through check valve 14, pipe 15, unloader valve 16 and pipe 3.

The path of the fluid is thus seen to be through a closed circuit to which are connected on both the high and the low pressure sides reservoirs which serve to prevent any sudden changes in pressure. The engine supplies the power to the compressor which is used to pump fluid, usually air, into the high pressure reservoir where the used energy is stored as potential energy of the fluid. This energy is available for driving the fluid motor which again transforms it into mechanical energy at the wheels. The closed circuit in a fluid transmission is particularly advantageous because of the huge outputs of power that can be transmitted with relatively small equipment, especially when the ratio of the pressures on the high and low pressure sides of the system is made about 3. Another advantage resides in the fact that, since the compression ratio is small, the temperature of the discharged air from the compressor is also low, and for the same reason, there is no freezing at the exhaust of the fluid motor.

For best efficiency it is desirable to maintain the fluid pressure in the high pressure side of the system substantially constant. To this end the speed of the internal combustion engine is varied to correspond to the demand for fluid and when the speed is reduced to a little above the stalling speed the compressor is unloaded. Fluid pressure reaches diaphragm pressure regulator 18 by way of pipes 6 and 19 from high pressure reservoir 4. The regulator 18 causes rod 20 to move in a direction to gradually close throttle 21 of carburetor 22 through linkwork 23 when the pressure in reservoir 4 increases. The effect of this is to decrease the speed of the engine and compressor and decrease the amount of fluid pumped into the reservoir as pressure increases. When a pressure is reached at which a speed near to stalling occurs, the rod 20 is advanced to a point where the valve 16 is closed, shutting off the intake supply to the compressor 2 which normally would be through pipe 3, valve 16, pipe 15, check valve 14, and pipe 12 from reservoir 13. After the intake is shut off, the compressor does no work and the engine idles. This condition persists until the pressure in reservoir 4 drops sufficiently to permit the valve 16 and the throttle 21 to open whereupon pumping again starts. If, for any reason, the pressure regulation fails and the pressure exceeds the safe limit, it is discharged from the high pressure reservoir 4 through pipe 6, the safety relief valve 24, pipe 25, and pipe 12 into the low pressure reservoir 13.

In order to make up for leaks of fluid from the system into the atmosphere, an automatic make-up means is provided. This can be seen in detail by referring to Fig. 2. A compressor 26 is mounted on the frame 17 and has a shaft which is coupled to the shaft of engine 1 so that the compressor runs whenever the engine runs. 27 is the intake pipe and 28 is the discharge pipe. In the particular embodiment illustrated, air is employed as the working fluid, so in the operation of the make-up means, when more air is demanded by the system, air is taken into the compressor 26 through cleaner 29, pipe 30, valve 31, pipe 27 and its associated compressor check valve and discharged through pipe 28 and its associated compressor check valve and pipe 32 into pipe 15

where it can be taken in by compressor 2 and pumped up into the high pressure side of the system. The air cannot, however, pass from pipe 15 back into the low pressure reservoir 13 because of the check valve 14 which permits fluid to pass only in a direction away from pipe 12.

The operation of the make-up system is automatically controlled. When the pressure in pipe 15 exceeds a predetermined value, it acts through pipe 32 on diaphragm regulator 33 to close valve 31 which shuts off the intake to the compressor 26, unloading it and causing it to stop discharging air into pipe 15. In the operation of the regulator 33, pressure acts on diaphragm 34 against the compression of spring 35, the reaction of which is taken by the bracket 36 mounted on the frame 17, to move the rod 37 through packing 38 in a direction to bring the valve cone 39 against the valve seat 40. Should the pressure regulator 33 not function properly and the pressure in pipes 15, 32, and 28 rise to a dangerously high value, the safety valve 41 opens and passes fluid out through pipe 42 to the atmosphere.

The control of the motion of the car is through two levers: lever 43 which controls throttle 8 in the high pressure line to the fluid motor 10, and lever 44 which, through linkwork 45, controls the valves of motor 10. The fluid motor 10 is like an ordinary steam engine having cylinder 46, piston 47 on piston rod 48, driving crank 50 through connecting rod 49. Crank 50 is on axle 51 attached to wheels 11. The valve mechanism 52 controls the time of cut-off, admission and exhaust and, in addition to being reversible, is capable of variation over a wide range of cut-off. The valve events are, of course, determined by the phase relation of valve 53 with respect to piston 47, which relation is changed by manipulation of the mechanism 52 through movement of the lever 44. A standard Stephenson link has been shown and since this valve gear is so old and so well known to those skilled in the art, it is not considered necessary to explain its operation in detail.

In the normal operation of the car in the forward direction, fluid pressures in the high and low pressure reservoirs stay substantially constant. The throttle 8 is wide open; the valves of the fluid motor are set in the forward position with the desired cut-off; the fluid passes from the high pressure side of the system through the fluid motor to the low pressure side from which it is compressed into the high pressure side again by compressor 2. The compressor speed varies and the compression starts and stops in response to the pressure in the reservoir 4. The compressor 26 compresses and idles at intervals depending upon the need for make-up fluid as shown by the pressure in pipe 15.

One of the most important advantages of the type of transmission here described is that it can be employed as a brake for the vehicle which it drives. This is accomplished by setting the valves of the motor in reverse which makes a compressor of the motor. Under these circumstances fluid is taken from low pressure reservoir 13 through pipe 12 into the motor 10 and is compressed and sent out through pipe 9. The energy required for the compression is supplied by the car in coming to rest. If the throttle 8 were left open during the braking operation, the car would immediately start backwards upon coming to rest, since the valves would be in reverse. Hence it is necessary to close the throttle 8. But it is also necessary during braking to provide a path from pipe 9 into the high pressure reservoir 4 and this path must not be conducting in the other direction. A check valve 54 in pipe 55 is, therefore, made to shunt the throttle 8. This check valve permits fluid to be pumped by motor 10 into the tank 4; but when the car has stopped, the fluid cannot return to the motor to propel the car in the reverse direction.

During braking, fluid is taken from the low pressure reservoir 13 and is pumped into the high pressure tank by motor 10. To prevent the lowering pressure in the tank 13 from causing the make-up apparatus to start working, the check valve 14 is placed in the pipe 15 between the low pressure reservoir and the pipe 32 leading to the make-up regulator 33 and the inlet to the compressor 2. This check valve 14 permits fluid to pass in the direction of the indicating arrow from the tank 13 and the pipe 12 into the pipe 15 and thence into the compressor intake and into the make-up regulator; so when the car is being driven by the transmission the fluid can circulate freely and if there is a real demand for more air in the system as evidenced by lowered pressure in the tank 13 it will quickly become apparent in pipe 15 when the compressor 2 starts working. When, however, the pressure in tank 13 is lowered not by leakage of fluid to the atmosphere but by having fluid pumped out of it in the braking process, the pressure in pipe 15 remains the same and, hence, there is no tendency for the make-up regulator to cause new air to be pumped into the system.

It is sometimes desirable to enlarge pipe 15 into a larger chamber as at 56 in order to make somewhat more constant the pressure in the pipe 15. Of course, chamber 56 may be simply a section of tank 13 with check valve 14 between the two sections.

What I claim as new and desire to secure by Letters Patent is:

1. In a closed pressure fluid transmission circuit including a compressor, a high pressure reservoir connected to the exhaust of said compressor, a reversible-valve fluid motor having an intake adapted to be connected to said high pressure reservoir, a low pressure reservoir connected to the exhaust of said fluid motor and adapted to be connected to the intake of said compressor, the combination of means automatically injecting fluid into the low pressure side of the circuit when the pressure in the intake pipe of said compressor is lower than a predetermined value, and valve means located in said circuit between said low pressure reservoir and the intake pipe of said compressor and adapted to prevent a lowered pressure in the low pressure reservoir from being communicated to the compressor intake pipe to start the automatic injecting means.

2. In a closed pressure fluid transmission circuit including a compressor, a high pressure reservoir connected to the exhaust of said compressor, a reversible-valve fluid motor having an intake adapted to be connected to said high pressure reservoir, a low pressure reservoir connected to the exhaust of said fluid motor and adapted to be connected to the intake of said compressor, the combination of means automatically injecting fluid into the low pressure side of the circuit in response to a pressure in the intake pipe of said compressor lower than a predetermined value, and a check valve in the compressor intake pipe between the low pressure reservoir and the point to the pressure of which the automatic injecting means is responsive, said check valve permitting

fluid to flow only in a direction from said low pressure reservoir toward said point.

3. In a closed pressure fluid transmission circuit including a compressor, a high pressure reservoir connected to the exhaust of said compressor, a reversible-valve fluid motor having an intake adapted to be connected to said high pressure reservoir, a low pressure reservoir connected to the exhaust of said fluid motor and adapted to be connected to the intake of said compressor, the combination of means automatically injecting fluid into the intake pipe of said compressor in response to a pressure in the intake pipe of said compressor lower than a predetermined value, and a check valve in the compressor intake pipe between the low pressure reservoir and the point to the pressure of which the automatic injecting means is responsive, said check valve permitting fluid to flow only in a direction from said low pressure reservoir toward said point.

4. In a closed pressure fluid transmission circuit including a compressor, a high pressure reservoir connected to the exhaust of said compressor, a reversible-valve fluid motor having an intake adapted to be connected to said high pressure reservoir, a low pressure reservoir connected to the exhaust of said fluid motor and adapted to be connected to the intake of said compressor, the combination of a chamber in the intake pipe to said compressor, means automatically injecting fluid into said chamber in response to a pressure in said chamber lower than a predetermined value, and a check valve in said circuit between said low pressure reservoir and said chamber permitting fluid to pass from said low pressure reservoir into said chamber but not to pass in the reverse direction.

5. In a closed pressure fluid transmission circuit including a compressor, a high pressure reservoir connected to the exhaust of said compressor, a reversible-valve fluid motor having an intake adapted to be connected to said high pressure reservoir, a low pressure reservoir connected to the exhaust of said fluid motor and adapted to be connected to the intake of said compressor, the combination of means automatically injecting fluid into the low pressure side of the circuit in response to a pressure in the compressor intake pipe lower than a predetermined value, and valve means located in said circuit between said low pressure reservoir and the intake pipe of said compressor and adapted to prevent the automatic injecting means from operating when fluid is exhausted from the low pressure reservoir by way of the fluid motor.

RUSSELL M. OTIS.

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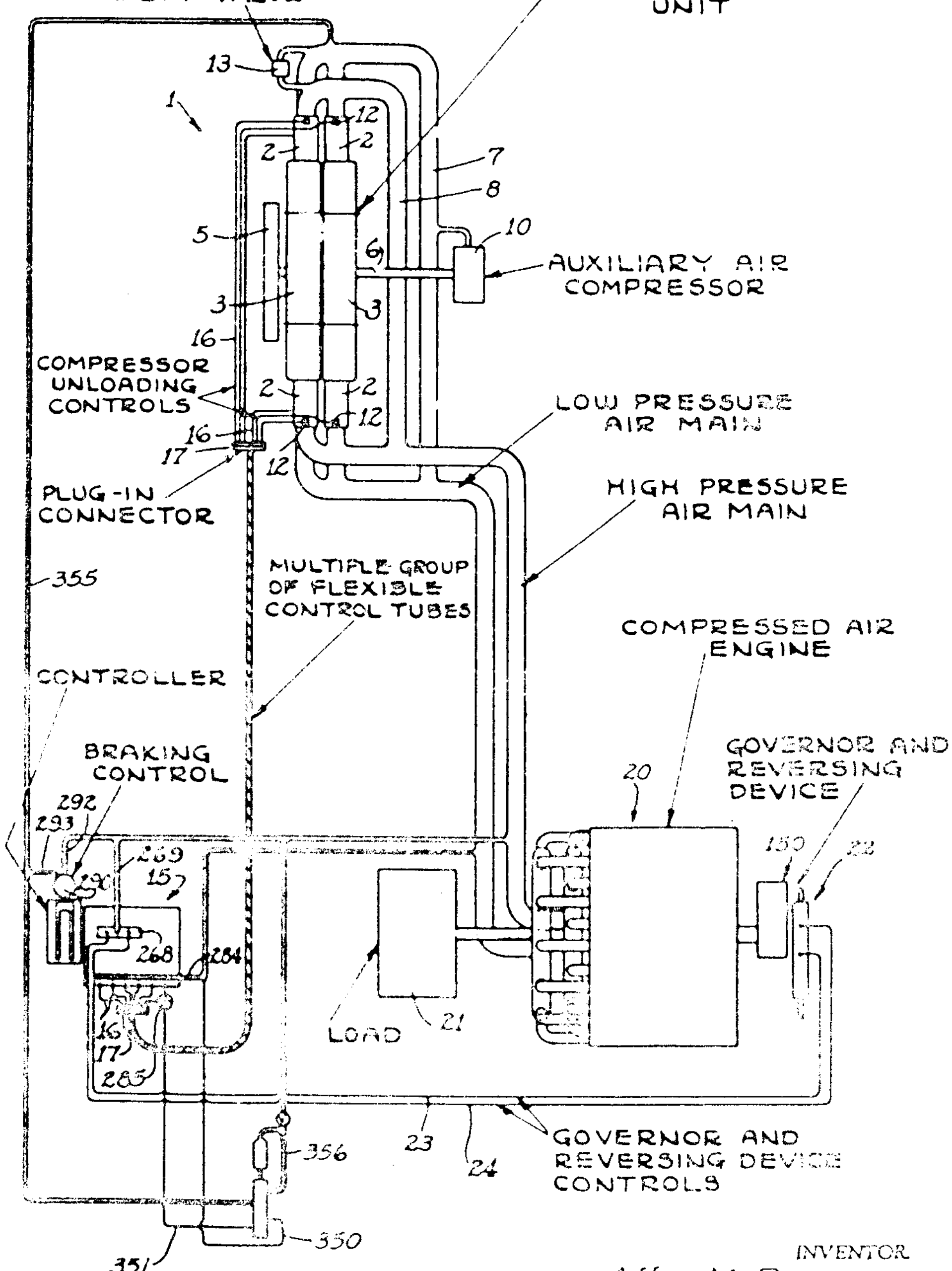
Fig. 1



INVENTOR
Allen M. Rossman
BY Morris Spector

Nov. 1, 1949 A. M. ROSSMAN 2,486,982

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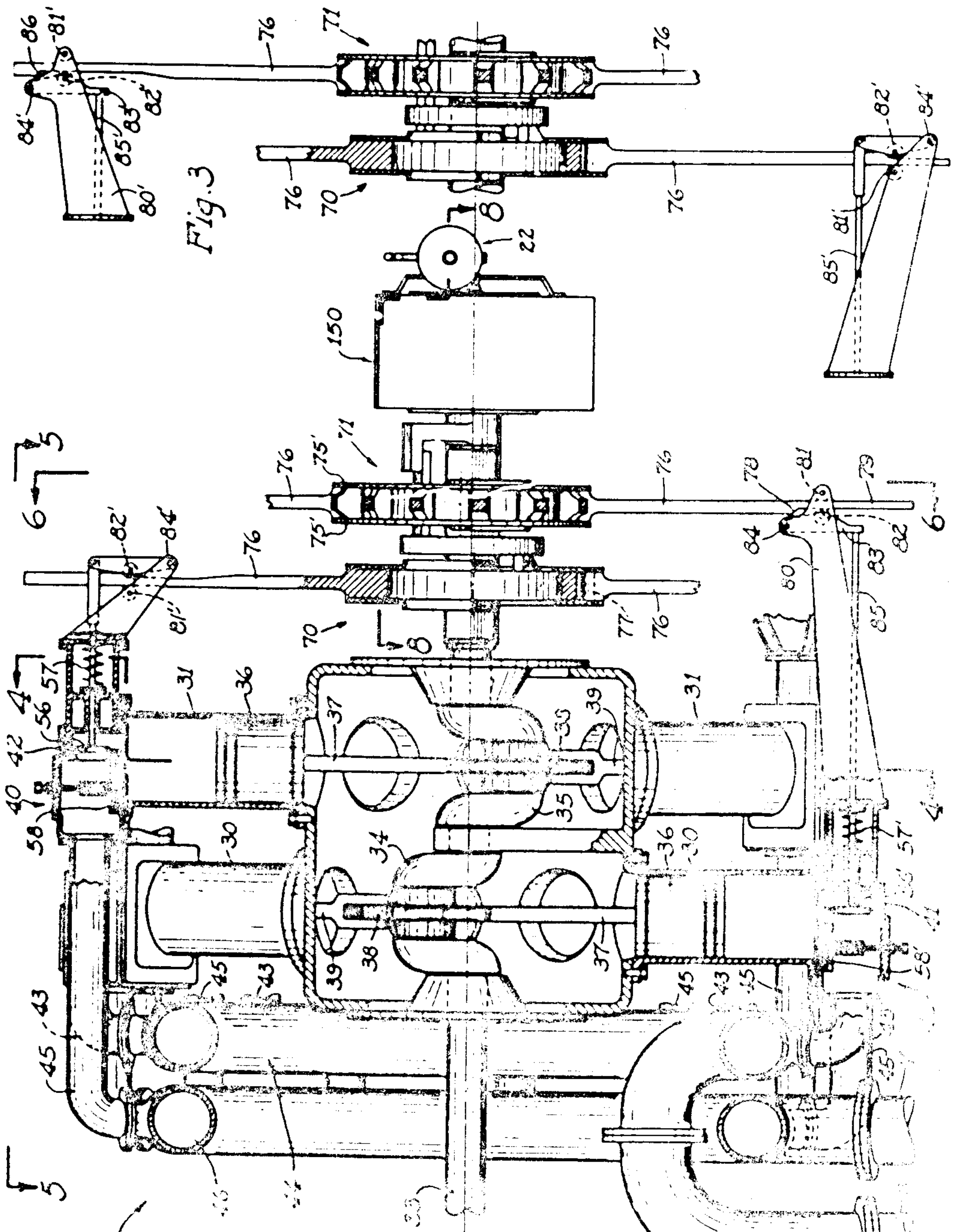

INVENTOR.
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PNEUMATIC POWER UNIT
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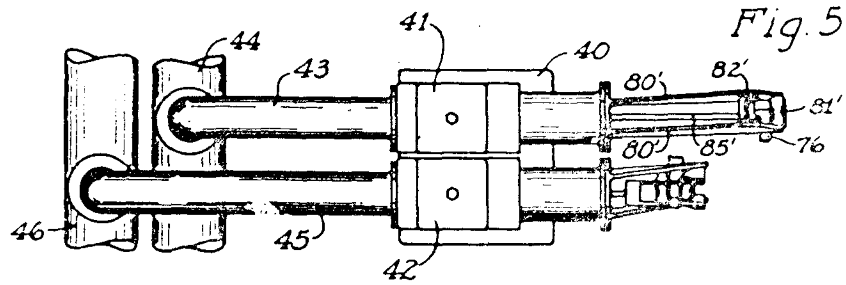

Fig. 5

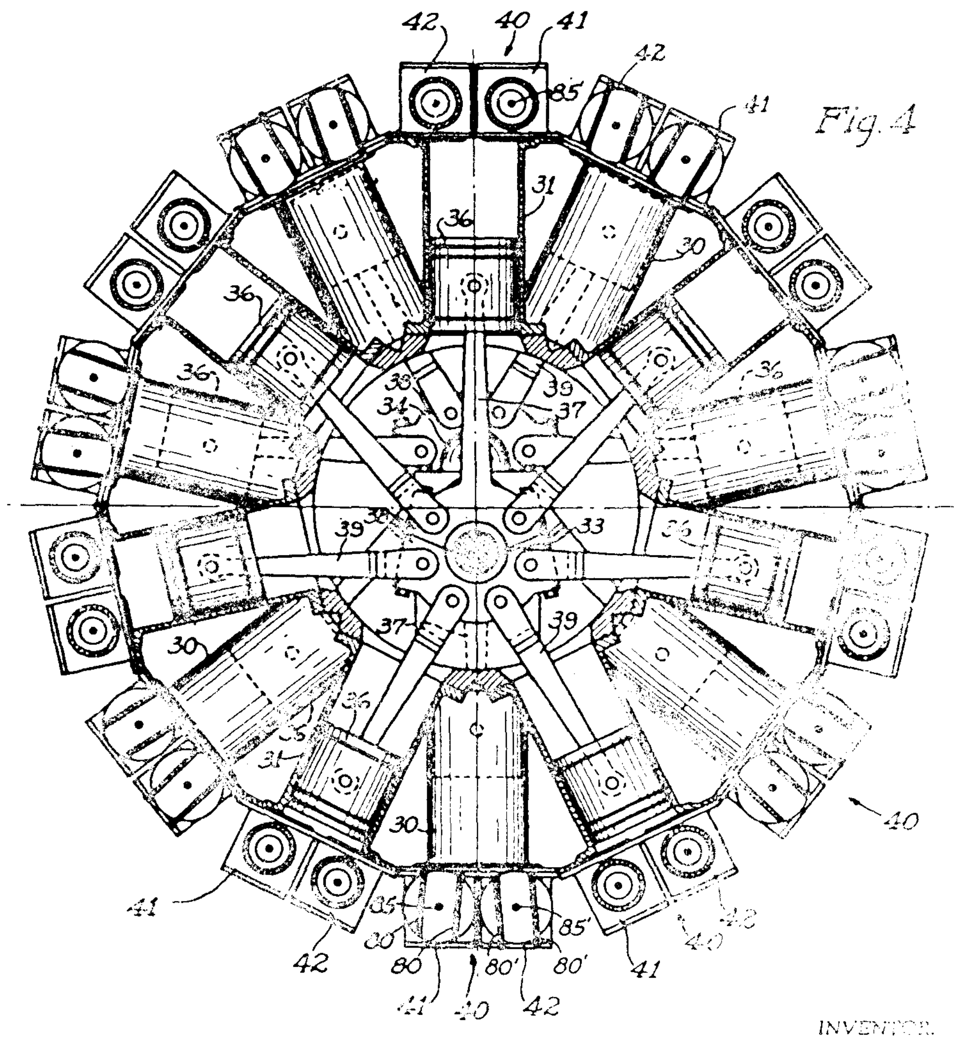

Fig. 4

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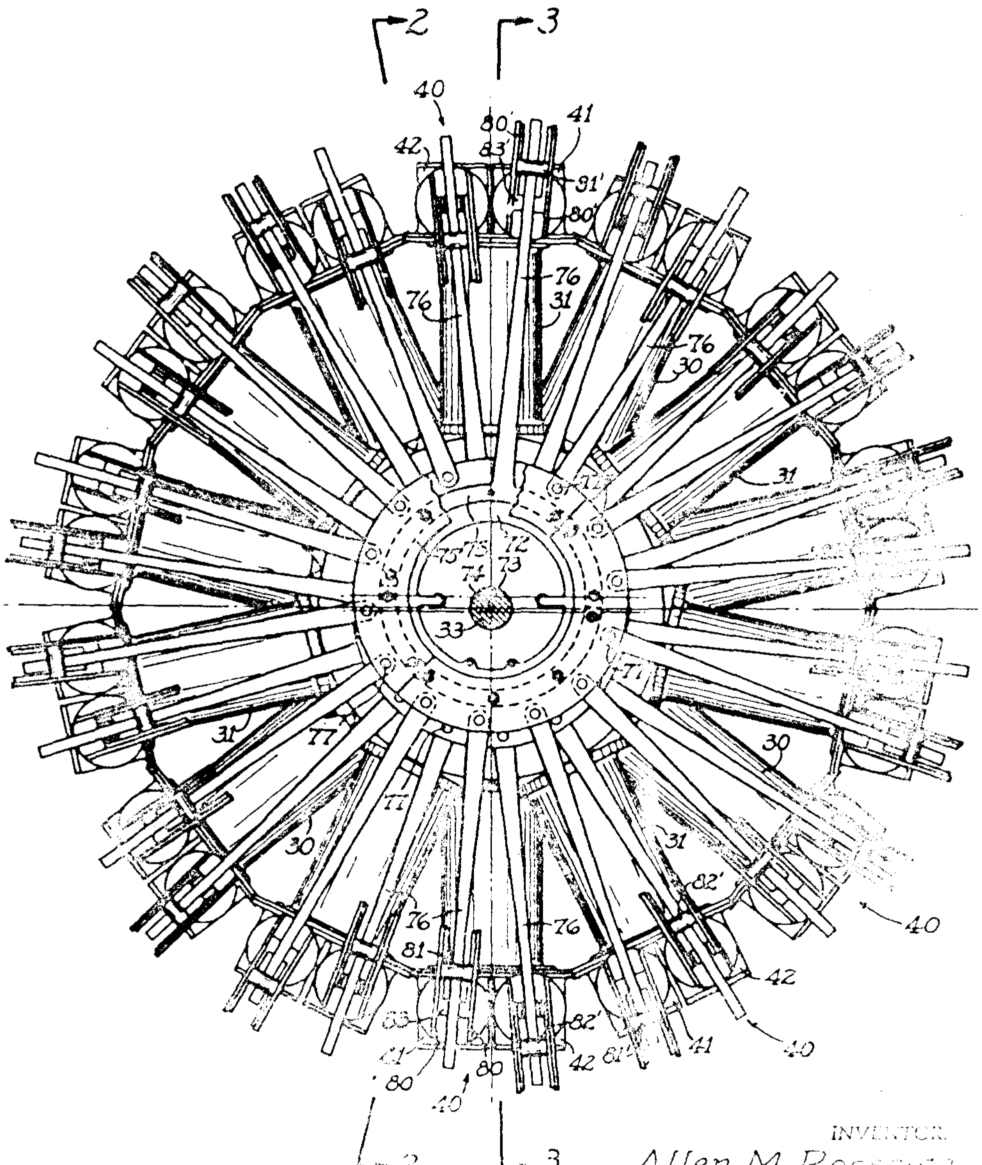
Fig. 6



INVENTOR.
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Morris Spector

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PNEUMATIC POWER UNIT
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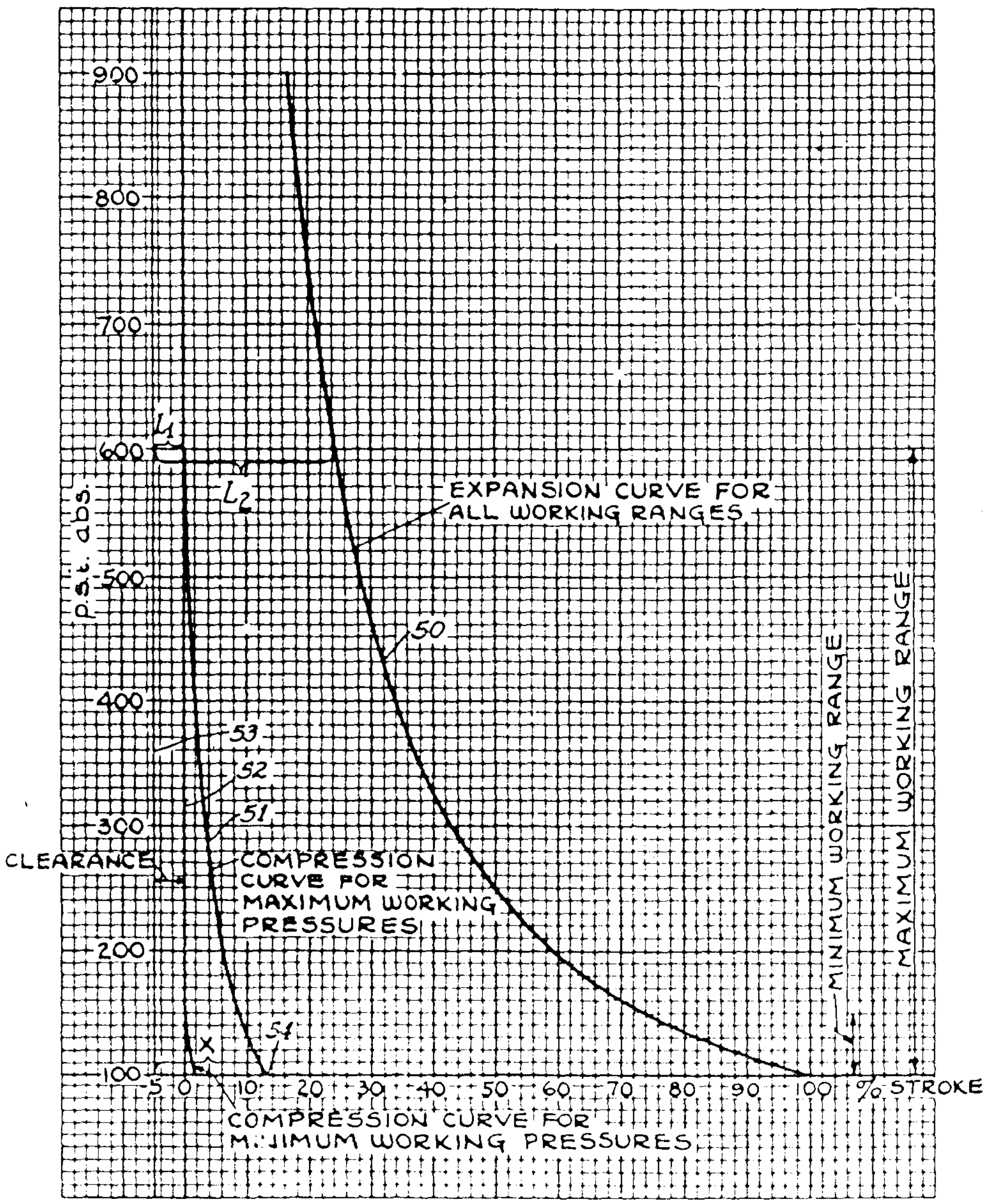
Fig. 7



INVENTOR.
Allen M. Rossman
BY Morris Spector

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PNEUMATIC POWER UNIT

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Fig. 8

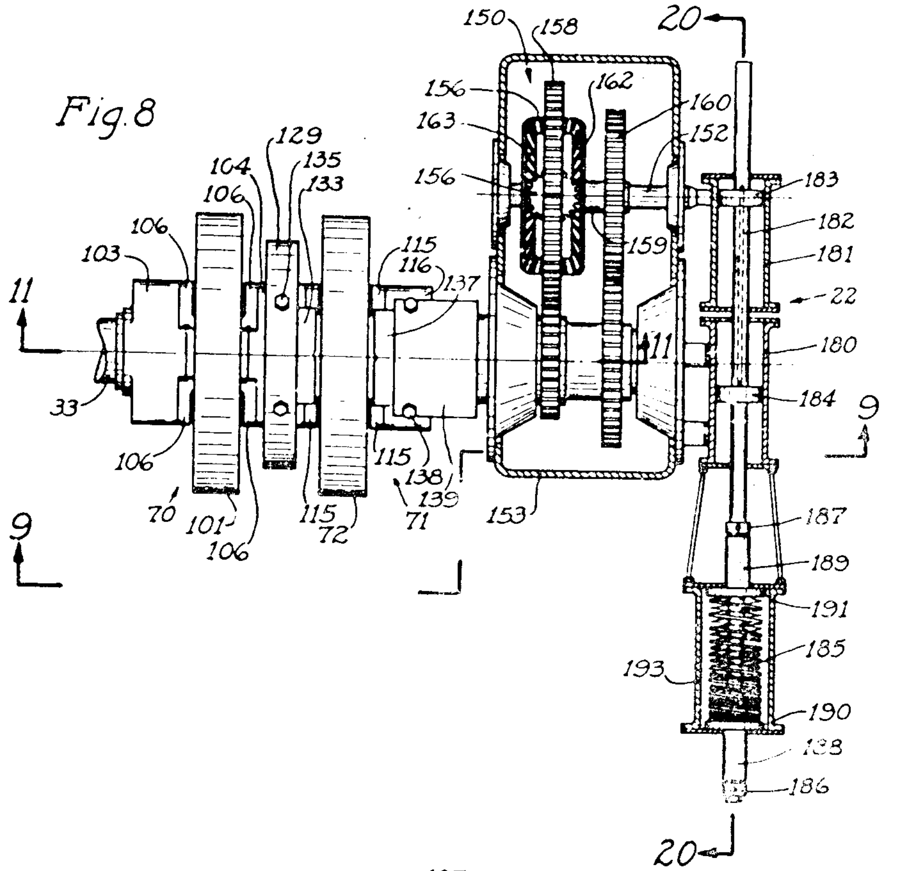
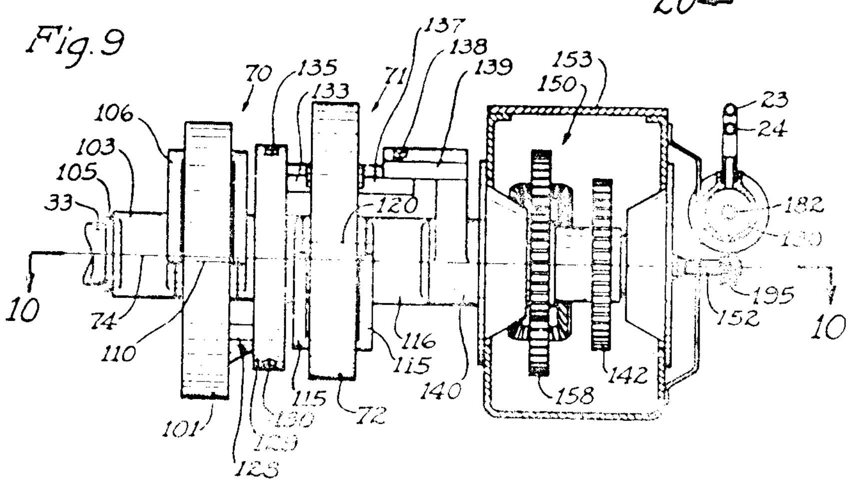


Fig. 9



INVENTOR.
Allen M. Rossman
BY Morton Spector

Nov. 1, 1949 A. M. ROSSMAN 2,486,982

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Fig. 20

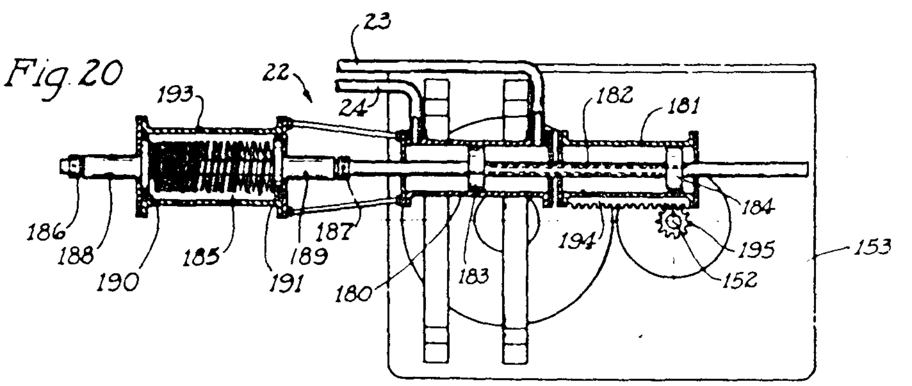


Fig. 10

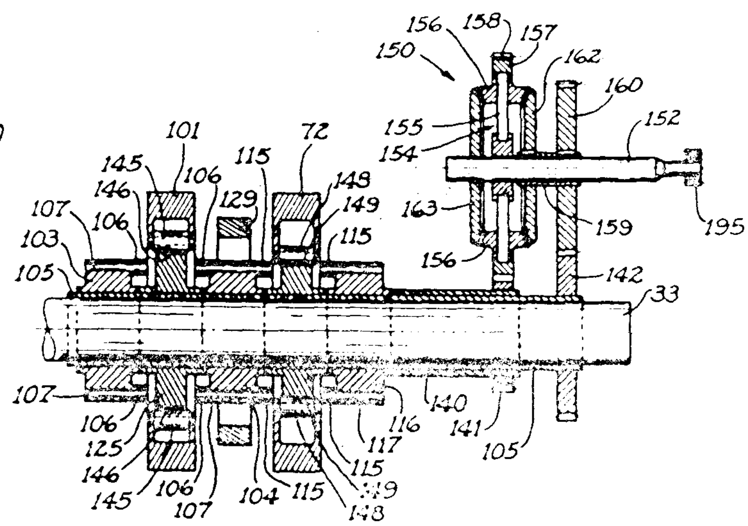
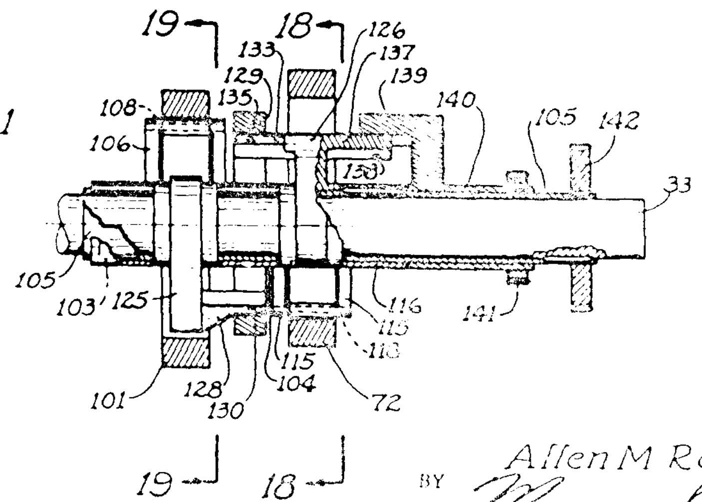


Fig. 11



INVENTOR.
Allen M Rossman
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Nov. 1, 1949 A. M. ROSSMAN 2,486,982

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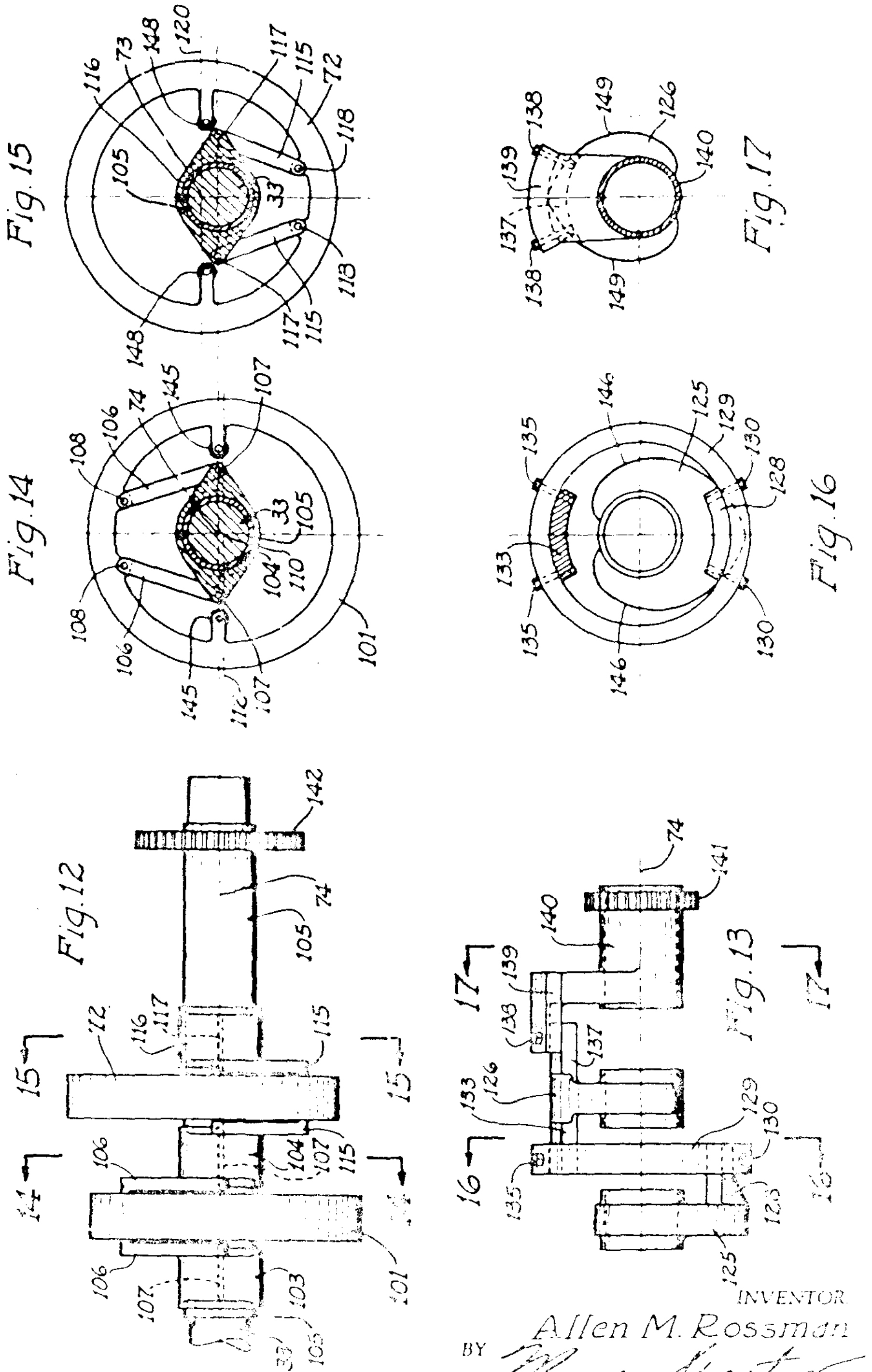

INVENTOR
Allen M. Rossman
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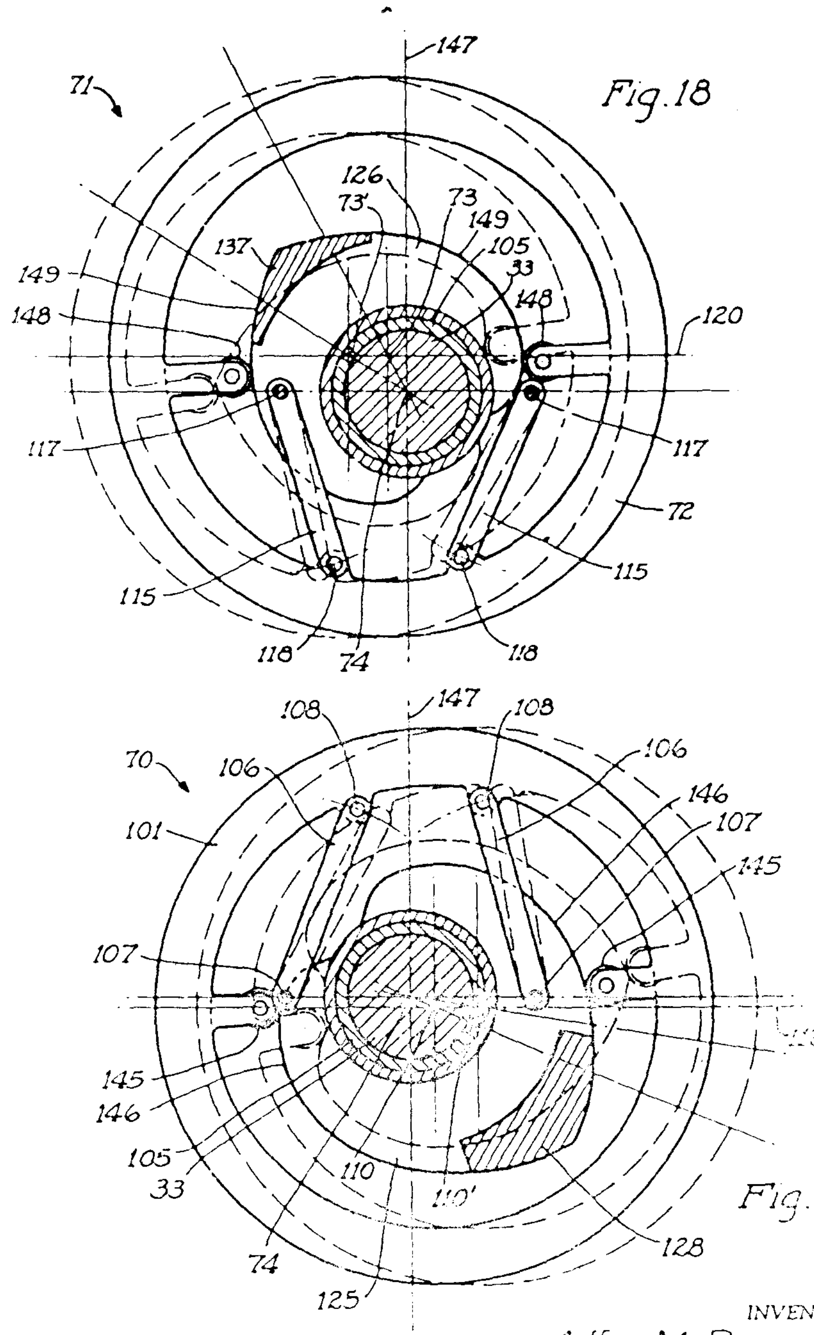

INVENTOR.
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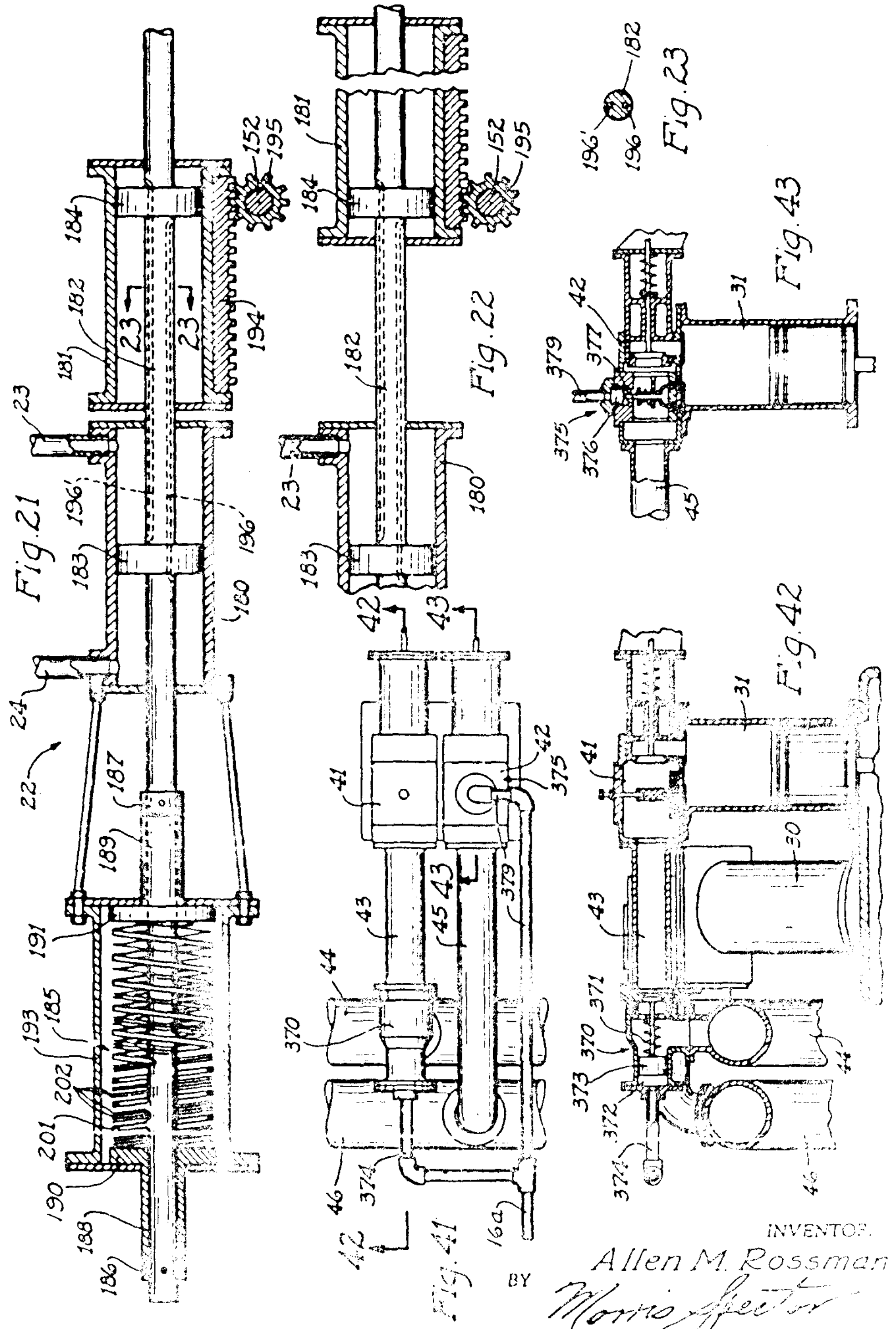

INVENTOR.
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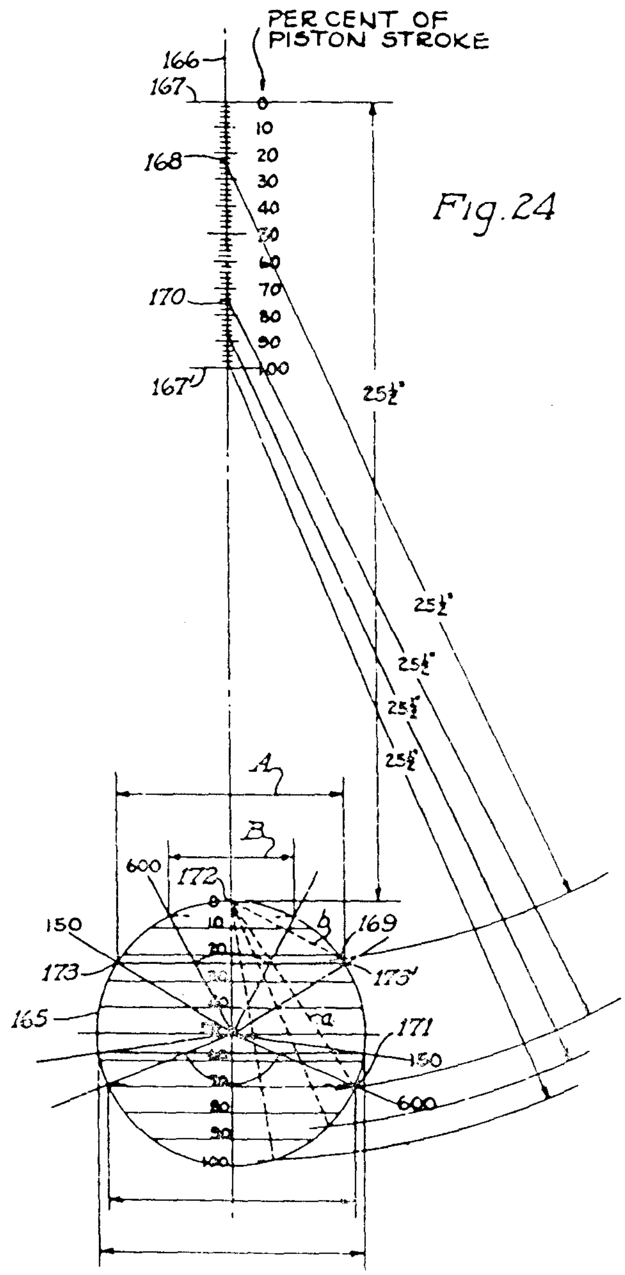

Fig. 24

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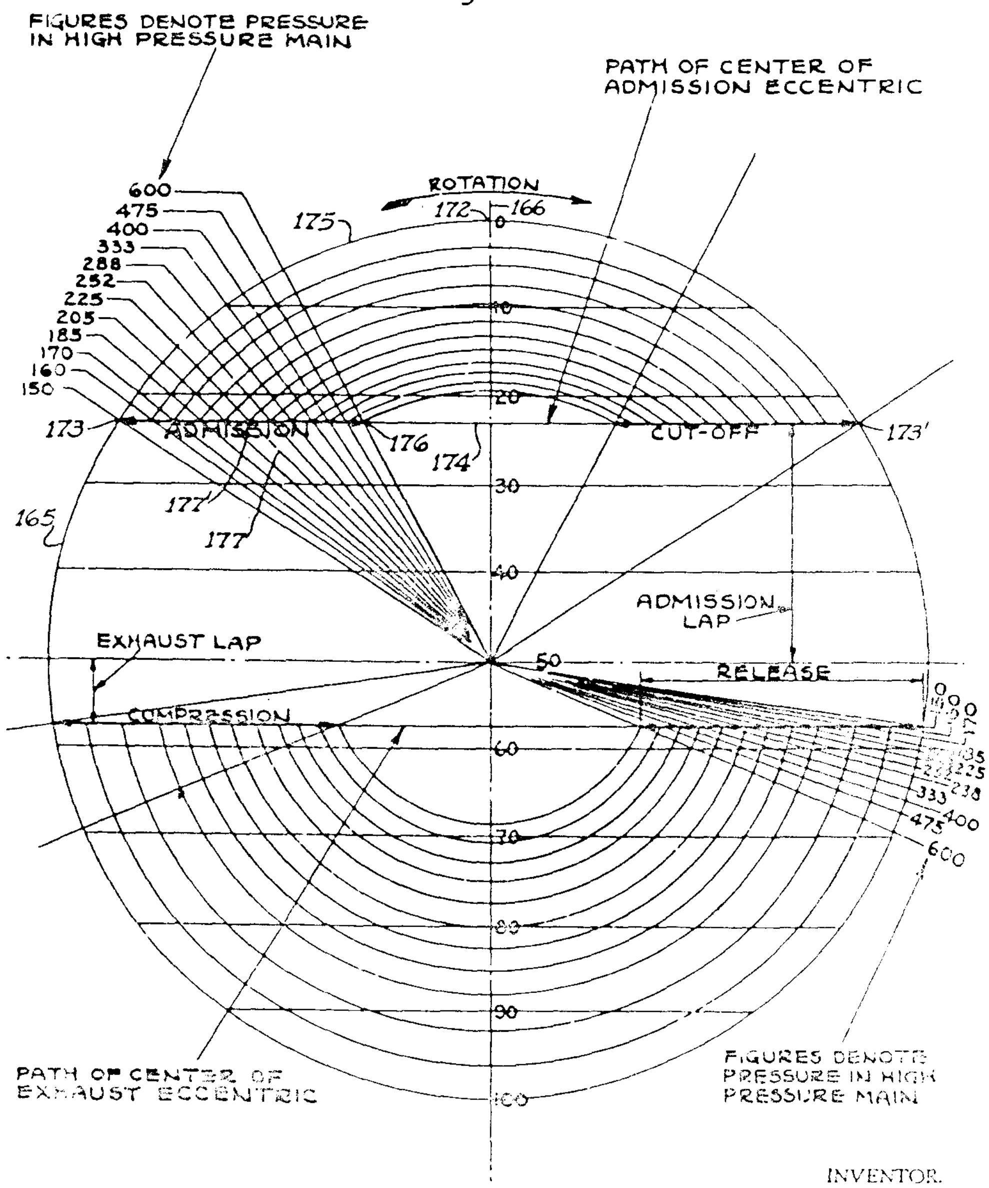
Fig. 25



INVENTOR.
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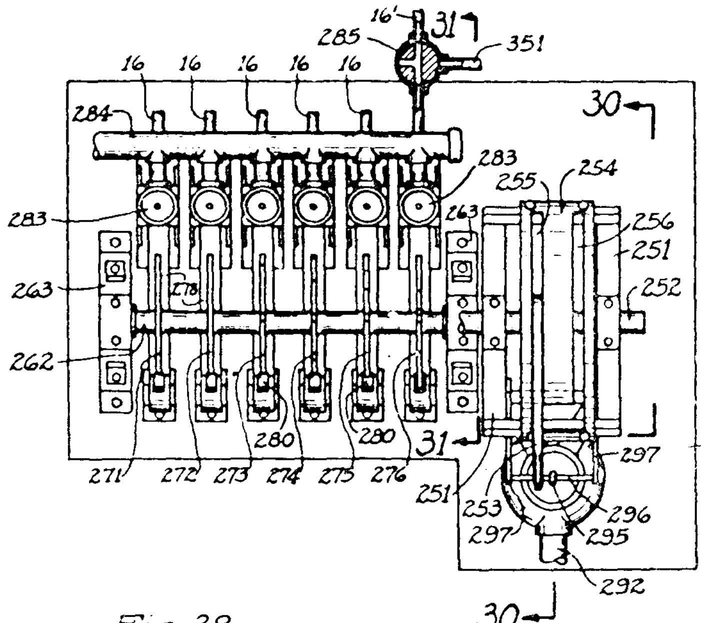
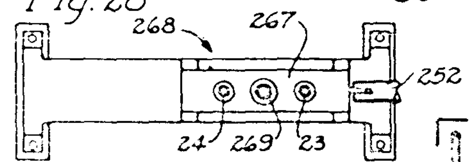


Fig. 26

Fig. 28



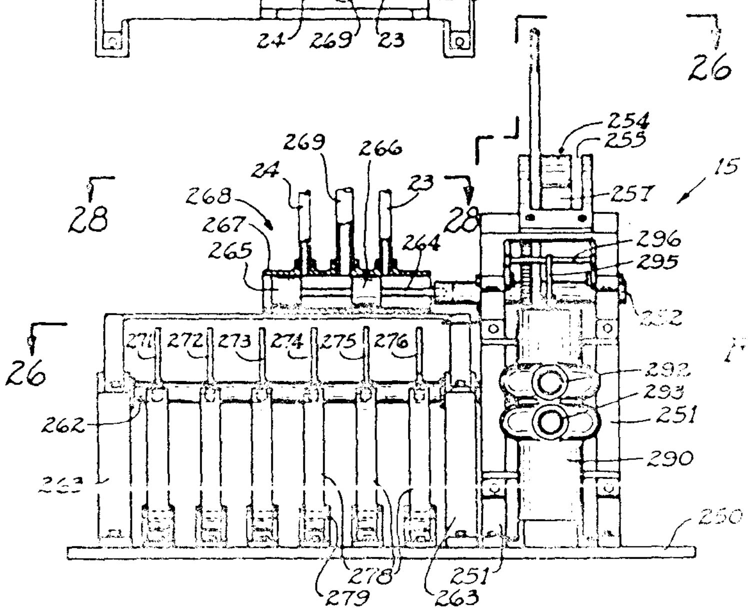


Fig. 27

INVENTOR.
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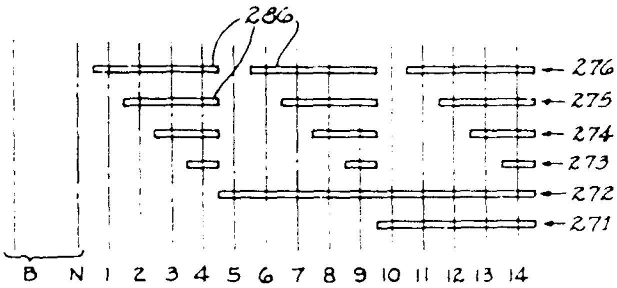

Fig. 33

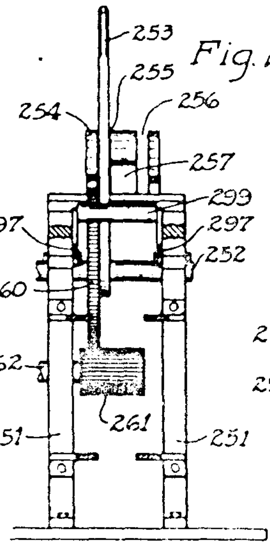

Fig. 29

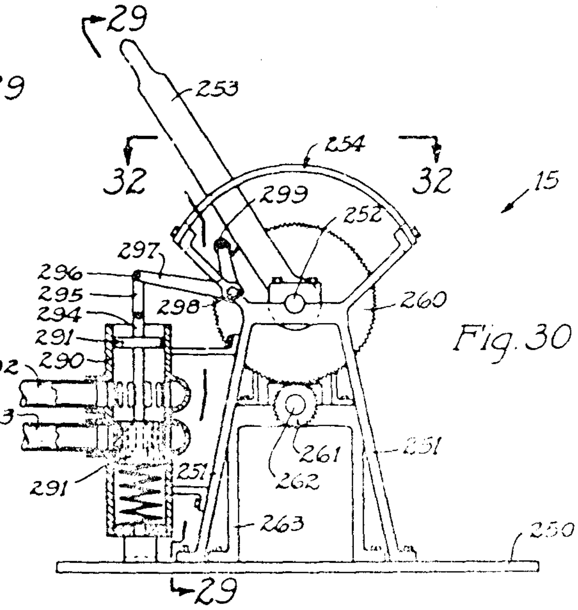

Fig. 30

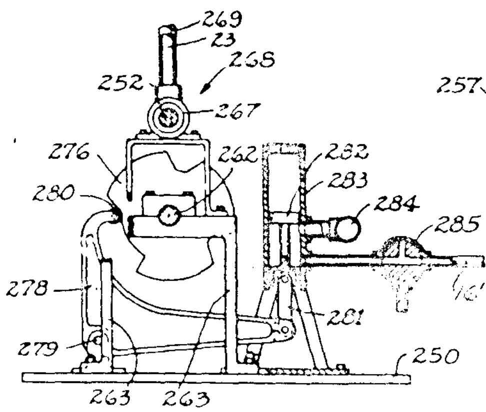

Fig. 31

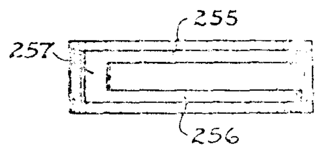

Fig. 32

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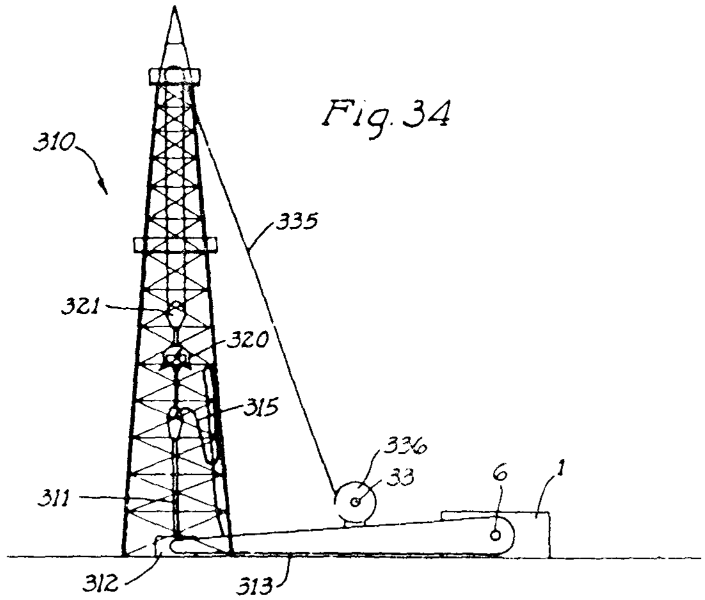

Fig. 34

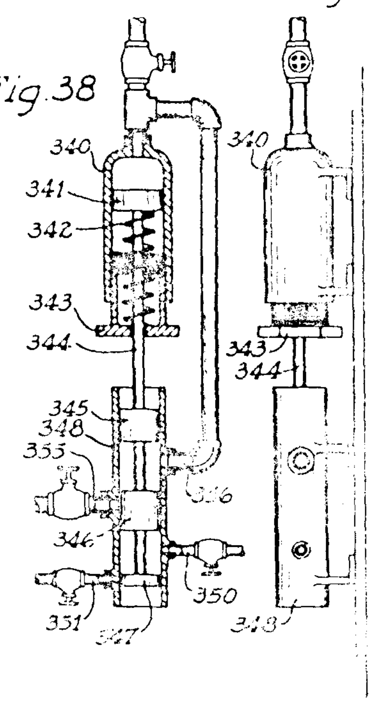

Fig. 38

Fig. 39

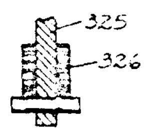

Fig. 37

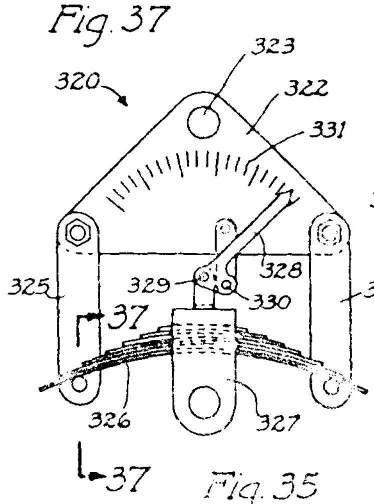

Fig. 35

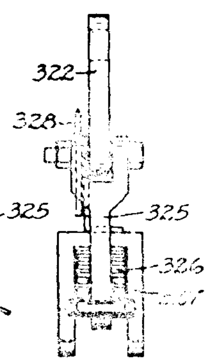

Fig. 36

INVENTOR

Allen M. Rossman

BY Morris Spector

Nov. 1, 1949 A. M. ROSSMAN 2,486,982

PNEUMATIC POWER UNIT

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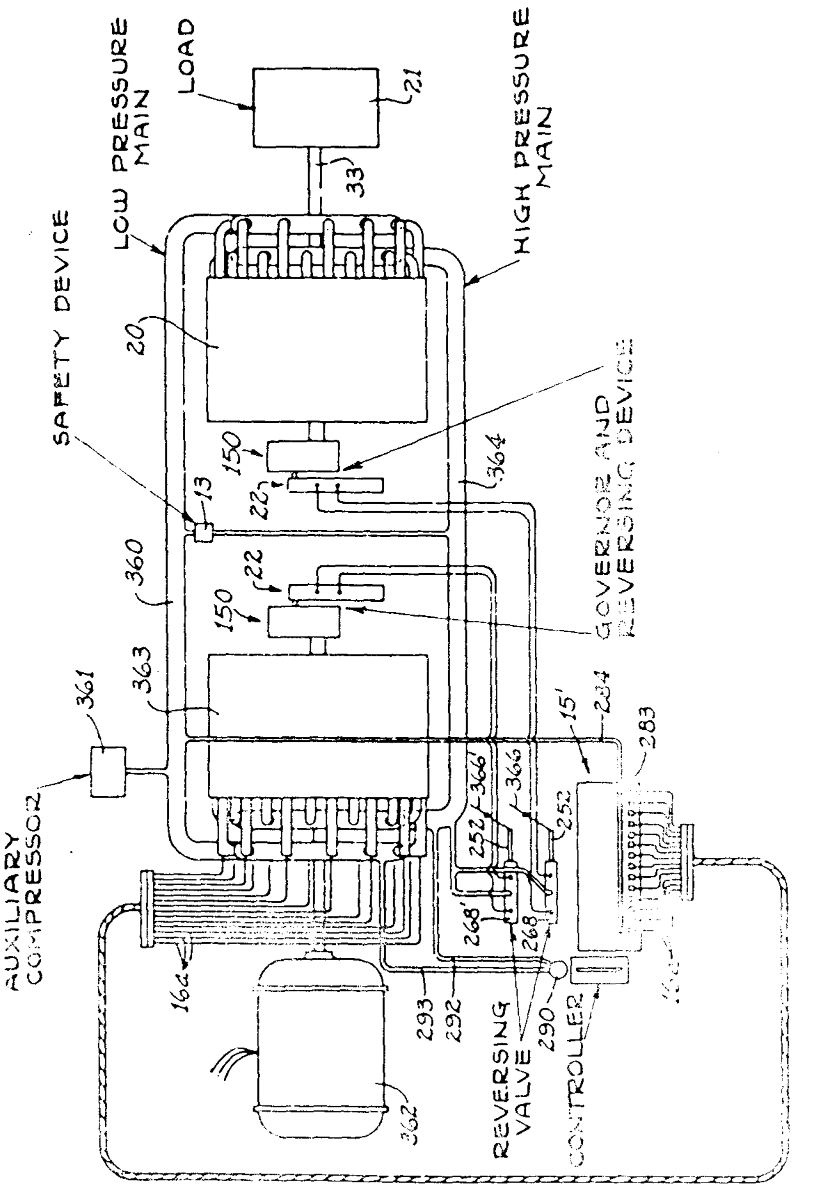
Fig. 40



INVENTOR.
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BY
Morris Spector

Patented Nov. 1, 1949 2,486,982

UNITED STATES PATENT OFFICE

2,486,982

PNEUMATIC POWER UNIT

Allen M. Rossman, Wilmette, Ill.

Application August 17, 1942, Serial No. 455,151

14 Claims. (Cl. 60—62)

1

This invention relates to power transmission systems and to equipment and machinery for such systems. While the system of the present invention is here illustrated as applied to an oil well drilling rig, it is to be understood that the invention is not limited thereto, the same being illustrative of one of the uses of this system.

In accordance with the preferred embodiment of the present invention the power system employs two main units, namely, an air compressor, and a compressed air engine, connected together by a high pressure main and a low pressure main. The air pressure system is a closed system operating between a low pressure which is preferably maintained constant, and a variable high pressure. The high pressure is maintained by an engine or other device which operates to deliver air to the high pressure main in such volume and at such pressures as the load demands. The pressure in the low pressure main is preferably kept at a constant value of several atmospheres, by an auxiliary compressor.

The air engine of the present invention takes air from a high pressure main at some pressure between fixed limits, which may be of the order of 150 pounds per square inch absolute to 600 pounds per square inch absolute, and exhausts it into a low pressure main at a constant pressure of, say, 100 pounds per square inch absolute.

As a result of the maintenance of an elevated pressure in the low pressure main the piston displacements of the air compressor and of the air motor may be greatly reduced. The reason for this is that for a given rating of power, speed and expansion ratio, the piston displacements vary inversely with the pressure in the low pressure main. For example, for absolute pressures in the low pressure main of 15 pounds per square inch and 100 pounds per square inch respectively, the corresponding piston displacements bear a ratio to each other of 100 to 15, respectively.

It is one of the objects of the present invention to provide a system of the above mentioned character wherein the air engine is located comparatively close to the compressor to enable it to take advantage of the rise in temperature of the air as it is compressed. In accordance with one of the features of the present invention the engine is so designed that an indicator diagram taken from any of its cylinders when the engine is operating at any load will be a close approximation of a diagram taken simultaneously from the air compressor. While the equipment is operating in this manner thermo-dynamic losses will be small because nearly all of the energy

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that is put into the air by the compressor, partly in the form of an increase in pressure and partly in the form of an increase of temperature, is recovered in the engine. Maximum and minimum designed pressure limits can be so chosen that maximum temperatures can be limited. With an initial temperature of 60° F. and a pressure range from 100 p. s. i. absolute to 600 p. s. i. absolute the temperature range under adiabatic conditions will be from 60° F. to 410° F. With temperatures of this order and with insulation judiciously applied, radiation losses can be kept low. There will be little tendency for a cumulative increase in the temperature of the air. Hence water jacketing of the cylinders is not required on either the compressor unit or the air engine unit.

The system of the present invention may use any multi-cylinder type of air compressor, the compressor being driven in any desired manner as, for instance, by an internal combustion engine or by an electric motor. The air intake valves of the air compressor are spring seated. Each air intake valve is provided with an unloading device which is pneumatically actuated to move the valve to its open position and to hold it in that position, so that reciprocation of the piston in that cylinder will result in no compression of the air therein, that piston merely idling.

When the load is at rest the air compressor is idled under control of its unloading devices. To start the load the unloading devices are progressively released, thus progressively bringing the compressor cylinders into action and gradually building up the pressure in the high pressure main. After the load has been started the pressure in the high pressure main is set to give the desired load speed. This may be done by manipulation of the unloading devices and, to some extent, if desired, by regulating the speed of the compressor. To reduce the load speed this process is reversed. During this speed cycle a governor is automatically at work on the compressed air engine making continuous adjustments of the cut-offs of both the admission valves and the exhaust valves to the end that an indicator diagram taken from any one of the engine cylinders may duplicate as closely as possible an indicator diagram taken simultaneously from a cylinder of the air compressor.

The method of control above described is based upon the following principles:

1. The torque developed by the air engine is

a direct and proportionate function of the mean effective pressure in its cylinders.

2. The mean effective pressure in the air engine cylinders is a direct, but not proportionate, function of the pressure in the high pressure main.

It follows from the above principles that there must be a value of pressure in the high pressure main that gives a mean effective pressure in the cylinders of the air engine that just balances the torque of the load. Any pressure above this value will cause the air engine to accelerate the load. Any pressure below this value will cause a load, such as a suspended weight, to drive the engine backward and thereby make the air engine function as an air compressor to brake the load. It is one of the objects of the present invention to provide a system wherein the mean effective pressure in the engine cylinders may be varied at will in order to make the air motor function in different manners as may be required for different operating conditions.

During the drilling of a deep oil well it is desirable that the weight or pressure exerted by the drill against the bottom of the hole in the ground formed thereby shall be maintained substantially constant. As the depth of the hole increases during drilling operations, the weight of the length of the drill pipe from the bottom of the hole to the top of the well increases. It is necessary that this increased weight shall not exert all of its pressure on the drill bit at the bottom of the hole. It is one of the objects of the present invention to provide a system wherein the compressed air engine that is used to furnish the power for raising or lowering the drill pipe may be used also, during the drilling operations, to hold a predetermined value of weight on the drill bit. During drilling, the air engine is set to exert a force tending to raise the string of pipe leading to the drill bit, but the air pressure is maintained at a value below that necessary to cause the engine to raise the pipe. If the torque exerted by the engine is maintained at a constant value it will allow a predetermined value of weight to be exerted on the drill bit. As the depth of the hole is appreciably increased, the pressure in the high pressure main is adjusted to a new value again to maintain a predetermined value of weight on the drill bit.

It is a further object of the present invention to provide an automatic mechanism for maintaining the pressure in the high pressure main at a predetermined set value during the drilling operations. It is a still further object of the present invention to provide such a mechanism which may be readily and quickly adjusted for maintaining different fixed values of pressure in the high pressure main, as may become necessary by the increase in length of the drill pipe during the drilling operation. It is a still further object of the present invention to provide a deep well drilling system with a device for indicating the amount of pull exerted by the engine on the travelling block that supports the drill pipe.

In accordance with one of the principles of the present invention the device for governing the air pressure in the high pressure main for maintaining a substantially constant weight on the drill bit is in the form of a relay which responds to the pressure in the high pressure main and builds up that pressure by releasing the unloading device on one or more cylinders of the air compressor when the pressure in the high pressure main is too low. On the other hand, should

the pressure in the high pressure main become too high the relay will shunt part of the air from the high pressure main into the low pressure main.

It is a still further object of the present invention to provide a power system with a controller so arranged that an air engine which drives the load may be used to brake the load in either direction of travel. While braking, the engine functions as an air compressor taking air from the low pressure main and discharging it into the high pressure main. It is another object of the present invention to provide a controller which not only controls the speed of operation of the engine but also controls the braking effort, regulating it to any desired value and for any speed within the designed limits.

When the engine is delivering power, the exhaust valve must open at the end of the power stroke and remain open during the major part of the subsequent return or exhaust stroke. Prior to the completion of that return stroke, the exhaust valve must close so that the air remaining in the cylinder is compressed to a pressure equal to that of the high pressure main, at which time the inlet valve is to open. An examination of adiabatic compression curves of air shows that because of the steepness of the slope of the curve it is impractical to make a mechanical valve setting close enough to effect opening of the inlet valve within the limits that appear desirable. To solve this problem, and for additional reasons, the engine is provided with two sets of inlet valves. One set is of the mushroom type, mechanically operated. The other set is of an automatically operated type that is spring closed when the pressure in the engine cylinder is below that of the high pressure main, and automatically opened by reversal of air pressure. This assures an opening of the high pressure air inlet valve at exactly the right time in the operation of the engine. The exhaust or outlet of each cylinder of the engine is also provided with an automatic valve which is spring closed and automatically opens when the pressure within the cylinder drops below that in the low pressure main, and with another valve also spring seated but mechanically opened for also controlling communication between the engine cylinder and the low pressure main. With this combination of valves, settings of the mechanically operated valves are so made that when the engine is hoisting, or driving its load, as the piston approaches the end of each stroke, either outwardly or inwardly, events will take place in the following sequence:

1. The automatic valve opens as soon as the pressure within the cylinder equals the pressure in the corresponding air main, and before the operation of the corresponding mechanical valve. This immediately equalizes the pressure on opposite faces of the corresponding mechanically operated valve.

2. The mechanically operated valve then opens.

3. The automatic valve then immediately closes but communication with the corresponding port is maintained by the mechanical valve and until the mechanical valve closes.

The mechanically operated admission valves and exhaust valves are operated by separate eccentrics. The governing action is obtained by moving the center of the eccentric with respect to the center of the operating shaft in a straight line at right angles to the crank, thus altering the eccentricity but keeping the lead constant. It is one of the objects of the present invention to pro-

vide an improved mounting for the eccentric on the drive shaft so that the eccentric center may be moved in a straight line. This is accomplished by mounting the eccentric ring on the shaft by means of a linkage which permits movement of the center of the eccentric ring with respect to the center of the shaft only in a straight line, and by providing a cam for effecting that movement. Ordinarily the cam and the eccentric ring move together with the drive shaft so that during the ordinary operation of the engine there is no relative movement between the eccentric ring and the cam. To shift the center of the eccentric ring with respect to the center of the shaft, the cam and the drive shaft are turned with respect to one another.

It is a still further object of the present invention to provide a mechanism for effecting relative motion between the eccentric ring and the cam that shifts the position of the eccentric ring with respect to the drive shaft while the drive shaft and the eccentric rings are in motion. It is another object of the present invention to provide a governor for governing the position of the center of the eccentric with respect to the drive shaft. The governor is actuated by pressure in the high pressure main and thus regulates the position of the center of the eccentric ring in accordance with changes in pressure in the high pressure main to maintain the action of the valves such that the expansion of the air in the air engine during the power stroke will be such that the pressure in the cylinder reaches its minimum value equal to the pressure in the low pressure main as the piston reaches the end of its power stroke.

It is a still further object of the present invention to provide a governing device of the above mentioned character which may be combined with a reversing device to set the engine valves for operation in either direction.

As the engine torque is increased, increments of air pressure increase at a faster rate than corresponding increments of torque. Compensation for this disproportion can be made in accordance with the present invention by incorporating in the governor design, a spring so proportioned that it will develop progressively increasing increments of counter-pressure, at the rate required to supply the proper degree of compensation.

It is a still further object of the present invention to provide a controller which can act through the reversing and governing mechanism to fix the direction of rotation of the air engine and which will also control the operation of the unloading devices on the air compressor in proper sequence and in such manner as to obtain the desired torque and speed from the engine; to change the function of the engine from motoring to braking; and to control the magnitude of the braking torque and hence the speed of the engine. It is a still further object of the present invention to provide a controller wherein the control of all of these functions is centered in a single operating lever, the position of which determines the function to be performed, and which may be moved to its various positions either by hand or by foot.

It is a further object of the present invention to provide an electropneumatic power system arranged to afford regenerative braking. The high pressure sides of two engines are connected together by the high pressure main and the low pressure sides of the engines are connected together by the low pressure main. One of the

engines is connected to operate the load whereas the other engine is connected to be driven by (or drive during braking) a constant speed alternating current motor. While the load is being driven the alternating current motor driven pneumatic unit operates as a compressor furnishing air to the other unit operating as a motor. During braking the functions of the two units are reversed so that the one connected to the alternating current motor drives the motor to feed energy back into the line. The electric driven unit is provided with pneumatically controlled unloading devices which are controlled by the manual controller to vary the output of the unit and thus vary the pressure in the high pressure main.

The attainment of the above and further objects of the present invention will be apparent from the following specification taken in conjunction with the accompanying drawings forming a part thereof.

In the drawings:

Figure 1 is a diagrammatic view of a power system embodying the principles of the present invention;

Figure 2 is a cross section through the compressed air engine of Figure 1, said view being taken along the line **2—2** of Figure 6;

Figure 3 is a view of a portion of the eccentric and valve rod structure of Figure 2 and showing parts of the valve rods that are omitted from Figure 2, said view being taken along the line **3—3** of Figure 6;

Figure 4 is a sectional view of the engine, said view being taken along the line **4—4** of Figure 2;

Figure 5 is a top view of a part of the compressed air engine of Figure 1, said view being taken along the line **5—5** of Figure 2 and looking in the direction of the arrows;

Figure 6 is a sectional view taken along the line **6—6** of Figure 2 and looking in the direction of the arrows;

Figure 7 shows adiabatic curves for air compression and expansion in the cylinders of the engine of Figure 2;

Figure 8 is a plan view of the valve operating eccentrics and of the actuating mechanism therefor, shown partially in section, said view being taken along the line **8—8** of Figure 2, with the cam operating rods omitted;

Figure 9 is a side elevation, in partial section, of the structure of Figure 8, said view being taken along the line **9—9** of Figure 8;

Figure 10 is a sectional view taken along the line **10—10** of Figure 9 and looking in the direction of the arrows;

Figure 11 is a sectional view taken along the line **11—11** of Figure 8 and looking in the direction of the arrows;

Figure 12 is a side elevational view of the eccentric rings, and the ring supports that are keyed to the main shaft;

Figure 13 is a side elevational view of the cams for adjusting the eccentric rings, and of the structure associated with the cams for securing the cams together;

Figure 14 is a sectional view taken along the line **14—14** of Figure 12 and showing the low pressure valve rod eccentric ring and the means for mounting the same;

Figure 15 is a sectional view taken along the line **15—15** of Figure 12 and showing the high pressure valve rod eccentric ring and the manner of mounting the same;

Figure 16 is a sectional view taken along the line 16—16 of Figure 13 and showing the cam for adjusting the low pressure valve rod eccentric ring;

Figure 17 is a sectional view taken along the line 17—17 of Figure 13 and showing the cam for adjusting the high pressure valve rod eccentric ring;

Figure 18 is an enlarged view taken along the line 18—18 of Figure 11 and showing alternate positions of the high pressure valve rod eccentric ring and the cam for adjusting it;

Figure 19 is an enlarged view taken along the line 19—19 of Figure 11 and showing, in dotted lines, an alternate position of the low pressure valve of the eccentric ring and the cam for adjusting it;

Figure 20 is a view taken along the line 20—20 of Figure 8 and looking in the direction of the arrows;

Figure 21 is an enlarged view of the reversing cylinder, governing cylinder and governing spring of Figure 20;

Figure 22 is a view of a portion of Figure 21 and showing the reversing cylinder in an alternate position;

Figure 23 is a sectional view taken along the line 23—23 of Figure 21;

Figure 24 is a valve diagram for the compressed air engine of the present invention;

Figure 25 is an enlarged and more complete view of a portion of the valve diagram of Figure 24;

Figure 26 is a plan view of the controller for the system of Figure 1, the pilot valve for the reversing mechanism being omitted, said view being taken along the line 26—26 of Figure 27;

Figure 27 is a front view of the controller of Figure 26, with the reversing pilot valve shown in section;

Figure 28 is a top view of the reversing pilot valve, said view being taken along the line 28—28 of Figure 27;

Figure 29 is a sectional view taken along the line 29—29 of Figure 30;

Figure 30 is an end view of the controller, with the brake control cylinder shown in section, said view being taken along the line 30—30 of Figure 26 and looking in the direction of the arrows;

Figure 31 is a sectional view taken along the line 31—31 of Figure 26 and looking in the direction of the arrows;

Figure 32 is a top view of the slotted guide plate for guiding the operating lever, said view being taken along the line 32—32 of Figure 30;

Figure 33 is a sequence diagram for the cam operation of the controller of Figure 26;

Figure 34 is a diagrammatic view illustrating the system of the present invention as applied to the draw works of an oil well drilling system;

Figure 35 is an enlarged front view of the drill bit weighing apparatus of Figure 34;

Figure 36 is an end view of the apparatus of Figure 35;

Figure 37 is a section taken along the line 37—37 of Figure 35;

Figure 38 is a sectional view through a relay for holding a predetermined value of weight on the drill bit in a deep well drilling apparatus;

Figure 39 is an end view of the relay of Figure 38;

Figure 40 is a diagrammatic view of a modified form of power system capable of regenerative braking;

Figure 41 is a fragmentary view corresponding to Figure 5 and showing the location of the pneumatically controlled valves on one of the power units of Figure 40;

Figure 42 is a sectional view taken along the line 42—42 of Figure 41 and looking in the direction of the arrows; and

Figure 43 is a sectional view taken along the line 43—43 of Figure 41 and looking in the direction of the arrows.

Throughout the various figures of the drawings like reference numerals designate like parts.

The power system is indicated in general in Figure 1, to which reference may be had. The system includes an air compressor unit 1 which may be of any desired construction, but in this instance consists of eight separate air compressor cylinders 2, each of a separate unit, said eight cylinders being driven in any desired manner as, for instance, by a motor or internal combustion engine 3. Ordinarily they are driven at a constant speed, and the outputs of the respective compressor cylinders 2 are controlled by unloader valves. The internal combustion engine includes a fly wheel 5 and a drive shaft 6. The compressor cylinders receive air from a low pressure main 7 and deliver air to a high pressure main 8. The pressure in the low pressure main is maintained at a constant value, preferably of several atmospheres, say, 100 pounds per square inch absolute, by an auxiliary air compressor 10 driven by the drive shaft 6. The compressor 10 takes air from the atmosphere and delivers it to the low pressure main 7 whenever the pressure therein drops below a predetermined value. Suitable automatic control devices, known in the art, are provided for controlling the output of the auxiliary air compressor 10 to maintain the pressure in the low pressure main at its predetermined constant value. As previously stated, the compressor cylinders 2 receive air from the low pressure main and compress it and deliver it to the high pressure main. Any one or more of the compressor cylinders 2 may be disabled by pneumatically controlling an unloading device 12 which maintains the air inlet valve to that compressor cylinder permanently open as long as the unloading device is actuated. Each unloading device 12 may be of any preferred design, one suitable design being shown and described in Marks' Mechanical Engineers Handbook, third edition, page 1373, to which reference may be had. The discharge valves of the compressor are spring seated and may be of any preferred design, suitable designs being illustrated and described in the same handbook on page 1866. A safety valve 13, normally closed, interconnects the high pressure and low pressure mains, and opens to interconnect those mains if the pressure in the high pressure mains becomes excessive.

While the load is at rest the air compressor idles under control of its unloading devices 12 which are controlled by a controller 15. The connections between the controller 15 and the air compressor, for controlling the unloading devices, are made of flexible metal tubing 16 which, for easy handling, can be grouped into a single cable with multiple plug-in devices 17—17 at both ends. The controller 12 may be manually operated to actuate any number of unloading devices 12, to disable the corresponding compressor cylinders 2, and thus reduce the output of the compressor, as will be more fully described hereinafter. If all of the unloading devices are actu-

ated, the compressor idles. The controller 15 also controls a reversing device on the air engine, to be more fully described as this description proceeds.

A compressed air engine is indicated at 20. The engine receives air from the high pressure main 8, expands it to the pressure of the air in the low pressure main 7, and discharges into the low pressure main. In this instance the engine is shown as driving a load indicated at 21. A governing and reversing device 22 controls the eccentric cams that operate air intake and air outlet valves of the compressed air engine. The governing and reversing device 22 is pneumatically controlled by the controller 15 through a forward air tube 23 and a reversing air tube 24.

While any standard type of steam engine can be designed to operate on compressed air and therefore any type of compressed air engine may be used in the system of Figure 1, the service for which the engine is to be used will usually favor some one particular type of engine. The engine that appears to be best adapted to the draw works of an oil well drilling rig is of the radial type. A description of one such suitable engine will be given, for which reference may be had to Figures 2 to 6 inclusive. The engine 20, illustrated more fully in Figures 2 to 6, is a fourteen cylinder engine having two rows of seven cylinders each, radially arranged, and a crank shaft with two throws 180° apart. One row of seven radially arranged cylinders is indicated at 30. The second row of seven radially arranged cylinders is indicated at 31. A power crank shaft 33 has two crank throws 34—35, 180° apart. The two crank throws are connected to the respective pistons 36 in any desired manner, as shown for instance in Figures 2 and 4. A connecting rod 37 is pivotally connected to one of the pistons 36 and has a crank head 38 consisting of two split parts bolted together and embracing a bearing on the crank. The remaining connecting rods 39 of the set of cylinders 31 are pivoted to the respective pistons and to the crank head 38.

Each cylinder 31 has a cylinder head 40 thereon which is divided into a high pressure valve chest or head 41 and a low pressure valve chest or head 42. The heads on each of the fourteen cylinders are of identical construction. Each of the fourteen high pressure valve heads is connected by a separate high pressure conduit 43 to a high pressure header 44, and each of the fourteen low pressure valve heads is connected by a low pressure conduit 45 to a low pressure header 46. The headers 44 and 46 are connected respectively to the high pressure main 8 and the low pressure main 7.

It is desired that the valve action of the air engine shall be such that an indicator diagram taken on any cylinder thereof will closely approximate an indicator diagram taken on any one of the compressor cylinders. While the compressed air engine is driving the load the length of cut-off of the admission valve should be so controlled that at the end of the power stroke the pressure in the cylinder will closely approximate the pressure in the low pressure main. Also, the point of closure of the exhaust valve should be so controlled that at the end of the reverse stroke the pressure in the clearance space will closely approximate the pressure in the high pressure main. Figure 7 shows adiabatic expansion and compression curves for air between a low pressure of 100 pounds per square inch absolute and a high pressure of 900 pounds

per square inch absolute. Assume that the engine is to operate at 600 pounds per square inch. Air should therefore be admitted to the cylinder at 600 pounds per square inch until the piston has travelled 24.2% of its stroke. This value is obtained from the curve 50 of Figure 7 at 600 pounds per square inch absolute. At that point in the stroke of the piston the admission valve must close, and thereafter the air in the cylinder is expanded along the curve 50 until the piston reaches the end of its stroke, at which time the pressure in the cylinder will be 100 pounds per square inch absolute. The exhaust valve must then open and remain open until the piston has returned 87% of its stroke, whereupon the exhaust valve closes. For the remainder of the stroke compression of the air remaining in the cylinder takes place. This compression takes place along a curve such as indicated at 51, and continues until a value of 600 pounds per square inch absolute is reached, whereupon the admission valve again opens.

Because of the steepness of the curve 51 at the higher values of pressure, which becomes almost vertical at 600 pounds pressure, it will be impractical to make mechanically operated valve settings close enough to bring the pressure in the cylinder within the limits that appear desirable at the time a mechanical valve is to open. To solve this problem, and for additional reasons that will appear as this specification proceeds, each cylinder is equipped with two sets of admission valves and two sets of exhaust valves. One admission and one exhaust valve of each cylinder is of the mechanically operated, mushroom type. The other exhaust valve and the other admission valve are of the type that are automatically operated by reversals of air pressure. With this combination of valves, settings of the mechanically operated valves should be made so that while the engine is driving the load, the automatic valve opens first and then the mechanical valve opens.

An explanation will now be given of the valve action whereby the above results are obtained. The low pressure valve head 42 is illustrated more fully at the top of Figure 2. The low pressure valve head 42 of the cylinder 31 includes a mushroom type mechanically operated valve 56 which is urged to its closed position by a spring 57 and when opened establishes communication between the low pressure conduit 45 and the cylinder 31, through the valve. A second valve 58, of the automatic type, is operated by reversals of air pressure and may be of a specific construction such as shown, for instance, in Marks' Mechanical Engineers Handbook, third edition, page 1866. It is sufficient here to state that this valve is maintained closed by a light spring and opens whenever the pressure in the cylinder 31 drops a trifle below the pressure in the low pressure main 45. The high pressure valve head of the cylinder 31 is of the same construction as the high pressure valve head of the cylinder 30, illustrated in section at the bottom of Figure [illegible], and includes a mushroom type mechanically actuated valve 56′ which is maintained closed by a spring, and an automatic valve 58′ similar to the valve 58, which is normally closed and is arranged to be automatically opened when and if the pressure in the cylinder 30 slightly exceeds the pressure in the high pressure header 44, to which the high pressure valve head 41 is connected. With this combination of valves, settings of the mechanically operated valves should be

made such that when the engine is driving the load, as the piston approaches the end of its power stroke, events will take place in the following order: The pressure within the cylinder 31 approaches and then equals the pressure in the low pressure air main. The automatic valve 58 thereupon opens and thus continues to equalize the pressure on opposite faces of the mechanical valve 56. An instant later the mechanically operated valve 56 opens so that the reclosure of the automatic valve 58 is of no effect. It thus follows that at the time of operation of the mechanical valve 56 the pressures on opposite faces of the mechanical valve are identical. The mechanical valve is maintained open until the piston has made a predetermined fraction of its return stroke, whereupon the mechanical valve closes and compression of the air in the cylinder 31 takes place. During compression the pressure on the mechanical valve 56 assists in holding that valve closed. The compression continues until the pressure within the cylinder equals the pressure in the high pressure main 8, whereupon the automatic high pressure valve 58' opens thereby continuing to equalize the pressure on opposite faces of the mechanically operated intake valve 56'. A moment after the opening of the automatic valve 58', the exact time interval being unimportant, the valve 56' opens to continue to admit air from the high pressure header 44 into the cylinder, even after the automatic valve 58' recloses. The intake of high pressure air into the cylinder thus continues entirely under the control of the mechanically operated valve 56'. The piston is then making its power stroke. When the piston reaches a predetermined position, determined by the setting of the governor as hereinafter explained, the mechanical admission valve 56' closes, and the piston continues its power stroke under the force of the expanding air in the cylinder. The point of cut-off of the mechanically operated admission valve is so correlated to the pressure in the high pressure main that at the time of closure of that valve the amount of air in the cylinder will be such as to be expanded to exactly the pressure in the low pressure main when the piston reaches the end of its power stroke.

An explanation of the mechanism for operating the valves 56—56' will now be given. All mechanically operated exhaust valves 56 are operated by a single eccentric 70 on the shaft 33. All mechanically operated admission valves 56' are operated by another eccentric 71 on the same shaft. The motion set up by the eccentric is carried to each valve stem through a radial rod. This is illustrated in Figures 2, 3 and 6. In Figure 6 the eccentric ring of the eccentric 71 is shown at 72. The center of this eccentric ring is indicated at 73 whereas the center of the crank shaft 33 is indicated at 74, the distance between the centers 73 and 74 being the eccentricity of the eccentric. A split ring 75 fits over the eccentric ring 72, said eccentric ring rolling in said split ring 75. The split ring 75 has one valve rod 76 rigid with half of the ring. A pair of arcuate plates forming a split ring 75' extend over one face of the ring 75 and the eccentric ring 72. A similar pair of arcuate plates forming a second split ring extend over the opposite face of the ring 75 and the eccentric ring 72. The plates 75'—75' are bolted together and to the split ring 75 and thus secure the two halves of the split ring 75 together and keep the split ring 75 from sliding axially off of the eccentric ring 72. Valve rods 76 are

pinned at one end between the ring plates 75'—75' by pivot pins 77.

The inlet valve rod 76 of Figure 2 has a cam surface 78 and a reduced width portion 79. That valve rod extends between a pair of arms 80—80 of a bracket secured to the high pressure valve head 41, and is guided between a roller 81 carried by the arms 80 and a second roller 82 on a rocker arm 83. The rocker arm 83 is pivoted to the bracket arms 80 about a pivot 84. The motion of the valve rod 76 guides the rocker arm 83 which through a push rod 85 opens the mechanically operated admission valve 56' against the action of a closing spring 57'. The principal function of the rocker arm is to step up the motion imparted by the eccentric rod 76 at the cam surface 78 thereof. In Figure 2 the rocker arm 83 has a one to two step up so that the push rod 85 moves twice as far as the motion of the roller 82 of the rocker arm 83.

With the cylinders arranged in two rows, as illustrated in Figure 2, alternate valve rods 76 that are actuated by the eccentric 71 operate the mechanically operated admission valves of the row of cylinders 30 and the intervening alternate valve rods 76 that are actuated by the same eccentric 71 operate the mechanical admission valves of the cylinders 31. Those valve rods 76 which operate the admission valves of the cylinders 30 are effective through the cam surfaces 78 to open the valves upon radial outward movement of the valve rods, as illustrated in Figure 2. The intervening rods 76, namely, those that operate the mechanical admission valves of the cylinders 31, are effective to open the valves upon radial inward movement of the valve rods, as illustrated in Figure 3. To accomplish this, the valve rods which operate the admission valves of the cylinders 31 are each provided with a cam surface 88 (Fig. 3) that actuates a rocker arm 83' mounted on a bracket 80' that is secured to the high pressure valve head 41 of the cylinder 31, as illustrated in Figure 5. Upon radial inward movement of the valve rod the cam surface 88 is effective to open the valve by pushing the valve rod 85' to the left. In the position illustrated in Figures 2 and 3 the eccentric 71 is in its uppermost position so that upon turning of the eccentric, the rod 76 of Figure 2 is moved downwardly and the rod 76 of Figure 3 is also moved downwardly. Thus the valve rods which control the admission valves of the cylinders 30 operate alternately with the valve rods that control the admission valves of the cylinders 31. The respective valve rods are arranged so that an outward movement of a rod to open an admission valve of a cylinder 30 is followed by an inward movement of another rod almost 180° away to operate a valve of a cylinder 31 that is located nearly 180° away from the cylinder whose valve was last operated.

The mechanically operated exhaust valves of the two rows of cylinders are operated by the eccentric 70 through valve rods similar to the rod 76. To that effect alternate valve rods actuated by the eccentric 70 open the mechanically operated exhaust valves of the cylinder 30 by a radial outward movement, as may be seen from Figure 3, whereas the intervening alternate valve rods actuated by the eccentric 70 open the exhaust valves of the cylinders 31 by a radial inward movement, as may be seen from Figure 2. The arrangements of the valve rods are such that the exhaust valve of one cylinder 30 is mechanically opened followed by the opening of an ex-

haust valve of a cylinder 31 almost 180° distant therefrom. The eccentric and valve rod structure for actuating the exhaust valves are similar to the corresponding structure for actuating the admission valves, as above described, differing therefrom essentially in that the shapes of the adjusting cams, to be presently described, for adjusting the eccentricities of the eccentric rings, are different because the required exhaust cut-off is different from the required admission cut-off, as will be pointed out in a discussion of the valve diagram as this description proceeds.

The speed of the engine is varied, as previously stated, by varying the rate of air input into the high pressure main, in a manner to be more fully set forth as this description proceeds. The pressure in the high pressure main varies with variations in torque but not in direct proportion thereto. In order that an indicator diagram of a cylinder of the engine shall follow as closely as possible a diagram taken simultaneously from a cylinder of the compressor it is necessary that the point of admission and cut-off of the air be changed as the pressure of the high pressure main 8 is changed. This result is obtained by moving the centers of the eccentrics 70 and 71 in a straight line at right angles to the crank shaft, thus altering both the eccentricity and the cut-off but keeping the lead constant.

An explanation will now be given of the means for moving the center of the eccentric. For this purpose reference should be had first to Figures 8 through 19 showing the manner of mounting the eccentric upon the crank shaft 33. In those figures the valve rod operating heads and associated parts surrounding the rings of the eccentrics 70 and 71 have been omitted. The eccentric 70 that controls the mechanically operated low pressure valves includes an eccentric ring 101 similar to the eccentric ring 72 of the eccentric 71. The two eccentrics are mounted on a sleeve 105 that is keyed to the shaft 33. The sleeve 105 is a length of tubing on which the eccentrics are assembled, which tubing and assembly may then be slipped on the end of the shaft 33. The eccentric ring 101 is supported on the sleeve 105 in the following manner: A pair of similar collars 103—104 are keyed to the sleeve 105 on opposite sides of the eccentric ring 101. The collars 103—104 have integral ears extending therefrom, as may be seen from Figures 14 and 15. A pair of identical links 106—106 are pivoted to the collar 104 about pivot pins 107—107 and a similar pair of links are similarly pivoted to the collar 103. At their opposite ends the links are pivoted to and support the eccentric ring 101 by pivots 108. The links 106 are of equal lengths, approximately equal to the radius from either pivot 108 to the center 110 of the eccentric ring 101. The distance between the pivots 107—107 is approximately twice the distance between the centers of the pivots 108—108. The distance between the pivots 107—107, and between the pivots 108—108, and the lengths of the links 106—106 are so proportioned that a point on the perpendicular bisector of a line joining the centers of the pivots 108—108 moves in approximately a straight line. This point is the center 110 of the eccentric ring 101. This is a well known type of straight line motion linkage known as the Roberts's straight-line motion, and is described on page 39 of a book entitled, "Experimental Mechanics," by A. Frederick Collins, published 1931 by D. Appleton & Co. If desired the straight line motion mechanism may be the proportions illustrated in my Patent No. 2,198,635, issued April 30, 1940, to which reference may be had.

In Figures 14 and 19 the center of the shaft 33 is indicated at 74. The center of the eccentric ring 101 is indicated at 110. If the links 106—106 are tilted with respect to the collars 103—104 upon which the links are mounted, the point 110 will move in a straight line the locus of which is indicated by the line 112. Thus as the center point 110 is caused to move along the line 112 the distance between the center point 110 and the center 74 of the shaft 33 changes, thus changing the eccentricity of the eccentric ring 101.

The eccentric ring 72 is supported on the shaft in much the same manner as is the eccentric ring 101, as is illustrated in Figures 12, 15 and 18. The eccentric ring 72 is supported by four identical links 115, two of which are pivoted at one end to the collar 104 and the other two of which are pivoted at one end to a collar 116 similar to the collar 104 and likewise keyed to the sleeve 105. The links 115 are pivoted to the collar 116 by pivot pins 117—117 and pivotally support the eccentric ring 72 by pivot pins 118. In the case of the eccentric ring 72 the proportions of the links 115—115, the distance between the pivots 118—118 and the distance between the pivots 117—117 are such that a point 73 on the perpendicular bisector of a line between the centers 118—118 moves on a straight line as the links are pivoted about their pivots 117—117. The proportions are such that the point 73 coincides with the center of the eccentric ring 72. The locus of the point 73 is indicated by the line 120. It is to be noted that as the center 73 of the eccentric ring 72 is shifted by tilting of the links 115, its distance from the center 74 increases, thus increasing the eccentricity of the eccentric.

A low pressure cam 125 is loosely mounted on the sleeve 105 and is located within the eccentric ring 101. A high pressure cam 126 is also loosely mounted on the sleeve 105 and is located within the cam ring 72. The cams 125 and 126 are rigidly connected together to move in unison. For this purpose the cam 125 has, integrally formed therewith, a projecting lug 128 which extends into and is bolted to a ring 129 (Figs. 13 and 14) by a pair of bolts 130—130. The ring 129 loosely surrounds the collar 104 that is keyed to the sleeve 105. The cam 126 has a projection 132 integral therewith which also extends into and is bolted to the ring 129, as by bolts 135—135. The cam 126 has an additional projection 137 which is bolted at 138 to a projection 139 on an arm that extends from a sleeve 140 loose on the sleeve 105. A spur gear 141 is keyed to the sleeve 140. Another spur gear 142 is keyed to the sleeve 105. By turning the spur gears 141 and 142 with respect to one another, thus turning the sleeve 140 with respect to the sleeve 105 the cams 125—126 are moved in unison with respect to the eccentric rings 101 and 72 respectively.

The eccentric ring 101 has a pair of roller cam followers 145—145, as illustrated in Figures 18 and 19. These cam followers ride along the cam surfaces 146—146 of the cam 125. Likewise the eccentric ring 72 has a pair of roller cam followers 148—148 that ride on the cam surfaces 149—149 of the cam 126. Ordinarily the sleeve 140 of the unitary structure of Figure 13 rotates in unison with the sleeve 105 of the unitary structure of Figure 11. Hence there is no relative

rotation between the roller cam followers 145 and their cam 125, and no relative rotation between the roller cam followers 148 and their cam 126. Thus during the time when the engine speed is not being regulated the centers of the eccentrics 72 and 101 rotate about the center 73 of the shaft, with an eccentricity determined by the setting of the two cams 125—126.

The settings of the two cams 125—126 may be changed simultaneously by turning the sleeve 140 with respect to the sleeve 105. If this is done then the cam 125 is rotated about the axis of the shaft 33 as a center and the cam 126 is likewise rotated about the same center. Actuation of the cam 125 causes the cam to act on the cam followers 145—145 to tilt the linkages 106—108 of the straight line motion mechanism thereby shifting the point 110, which is the center of the eccentric ring 101, along the line 112 (Fig. 19). The turning movement of the cam 126 causes this cam to act on the cam followers 148—148 to tilt the links 115—115 of that straight line motion mechanism, thereby producing a movement of the point 73, which is the center of the eccentric ring 72, along the straight line 120. It is thus apparent that by an angular movement of the sleeve 140 with respect to the sleeve 105 there is produced a straight line shifting of the centers of the admission eccentric ring 72 and the exhaust eccentric ring 101. This alters the eccentricity of each eccentric but keeps the leads constant.

Figures 18 and 19 show, in full lines, the positions of the cams 126 and 125 controlling the eccentric ring 72 and 101 when the pressure in the high pressure main is 600 pounds per square inch absolute. At that time the centers of the eccentric rings 72 and 101 are at 73 and 110 respectively. The cams are rotatable from the positions shown in full lines to the positions shown in dotted lines to vary the admission cut-off and compression cut-off as the pressure in the high pressure main changes or is changed. The dotted line positions of the cams and eccentric rings are the positions assumed when the pressure is 150 pounds per square inch absolute. As the pressure changes from the upper pressure to the lower pressure the turning of the cam 126 shifts the center of the eccentric ring 72 along the line 120 to the point 73'. At the same time the turning of the cam 125 shifts the center of the eccentric ring 101 along the line 112 from the point 110 to the point 110'. Those positions represent the extremes of movement of the centers of the eccentrics 72 and 101 while the engine is operating in one direction. To operate the engine in the reverse direction the cams 125 and 126 are rotated clockwise past the center line 147. As this rotation takes place the center of the eccentric 72 moves to the right along the line 120 past the center line 147. The adjustment of the cam 126 for different pressures moves the center of the eccentric 72 along the line 120 between two points on the opposite side of the center line 147 similar to the spacings of the points 73—73' shown. The action of the cam 125 is similar to that of the cam 126.

A differential gear 150, best shown in Figures 8, 9 and 10, is provided for producing controlled relative rotation of the sleeve 140 with respect to the sleeve 105 during the operation of the motor. One rotary element of the differential is in mesh with the gear 141 and another rotary element of the differential is driven by the gear 142, the gear ratios being such that the gears 141—142 are rotated at the same angular speeds when the third rotary element of the differential remains stationary.

Reference may now be had to Figures 8, 9 and 10 for an explanation of the action of the differential gear 150. The differential includes a shaft 152 journalled in suitable bearings in a casing 153. A spider 154 is rotatably mounted on the shaft 152 and includes four stud shafts 155 spaced 90° apart and each carrying a bevel gear 156 freely rotatable about the stud shafts. The four stud shafts are journalled inside a ring 157 which has spur gear teeth 158 on the outside thereof. The teeth 158 are in mesh with the teeth of the spur gear 141. A collar 159 is rotatable on the shaft 152 and has a spur gear 160 and a bevel gear 162 keyed thereto. The gear 160 is in mesh with the spur gear 142. The spur gear 160 through the collar 159 drives the bevel gear 162 which is in mesh with the four gears 156 on the spider 154. The gears 156 are in mesh with a bevel gear 163 which is keyed to the shaft 152. For an understanding of the operation of the differential assume that the shaft 152 is held stationary. The crank shaft 33 rotates the gear 142 which drives the gear 160, which in turn rotates the collar 159 and thus the gear 162. The gear 162 drives the four gears 156. Since the shaft 152 is assumed to be held stationary the gear 163 is also held stationary. Thus the rotation of the gear 162 causes the gears 156 to drive the spider 154 about the shaft 152 as a center and thus rotate the gear 157, which in turn drives the gear 141. The differential thus interconnects the gears 141 and 142. Since the gear 142 is keyed to the sleeve 105 which in turn is keyed to the shaft 33, it follows that the shaft 33 drives the gear 142 directly, and drives the gear 141 through the differential. The gear ratios are such that with the shaft 152 held stationary the driving ratio of the differential, between the gear 142 and the gear 141, is a one to one ratio. The gear 141 is driven at exactly the same angular speed as the speed of the gear 142. Thus as long as the shaft 152 is held stationary the cams 125 and 126 are rotated at exactly the same speed as the speed of rotation of the eccentric rings and there is no relative motion between the two. If it is desired to produce a shift or relative motion of the cams 126—126 with respect to the crank shaft 33, it is merely necessary to turn the shaft 152 the desired number of degrees. This may be done manually or automatically, while the shaft 33 is rotating or is stationary, by means of a reversing device and governor, to be presently described.

As previously stated, the valve action of the engine of the present invention is such that an indicator diagram taken from any cylinder thereof will closely approximate an indicator diagram taken on any cylinder of the compressor unit or, stated conversely, a curve of the expansion of the air in the engine cylinder will closely approximate an adiabatic expansion curve of air between the limits of the intake and discharge pressure of the cylinder. It is therefore essential that the valve action shall be such as to give such a diagram. Referring now to Figure 7, it may be seen that when the engine is operating between the pressures of 100 pounds and 600 pounds per square inch absolute the admission valve should remain open during the first 24% of the piston stroke and then close. Expansion will then take place in the engine cylinder along the curve 19 so that when the piston reaches its limit of stroke

the pressure within the cylinder will be 100 pounds. On the other hand, when the engine is operating between 100 and 500 pounds pressure absolute the admission valve should remain open for the first 28% of the stroke (which is the value of the abscissa of the curve 50 at 500 pounds pressure) in order that upon expansion of the air in the cylinder the pressure will equal 100 pounds at the end of the stroke. The valve action is designed to give a variable admission depending upon the pressure in the high pressure main in accordance with the curve 50 of Figure 7. The design is attained in the following manner, for which reference may be had to Figures 24 and 25: Assume that the crank circle is indicated at 165 and assume a length of connecting rod six times the length of the crank. The locus of the travel of the piston end of the connecting rod is indicated by the line 183 between the limits 167 and 167'. This has been calibrated in terms of percentage of stroke from zero to 100. When the pressure in the high pressure main is 600 pounds the admission port must be open for 24.2% of the stroke, a figure obtained from Figure 7, curve 50, at 600 pounds pressure. On the scale 166 a point 158 is located at a position equal to 24% of the piston stroke. With this point as a center and with the length of the connecting rod as a radius, an arc is struck intersecting the circle 165 at a point 169. At 150 pounds pressure the admission port must be open for 73.5% of the piston stroke, as may be determined from the curve 50 of Figure 7. Those are the limits of pressure at which the engine is to operate. A point 170 is located on the line 166 at 73.5% of the stroke. With that point as a center and with the length of the connecting rod as a radius an arc is struck which intersects the circle 165 at 171. The cord *a* extending from the dead center position 172 to the point 171 subtends an angle from the center of the circle 165 equal to the angle of travel of the crank required to give an admission opening during 73.5% of the piston stroke, whereas the similar cord *b* extending from the point 172 to the point 169 on the circle 165 subtends an angle equal to the angle of travel for the crank required for 24.2% cut-off. The angle that the cord *a* makes with radii at the opposite ends thereof is therefore laid off equally on opposite sides of the center line 166 so that the cord A equals the cord *a*. This establishes the points 173 and 173'. The line 174 adjoining the points 173—173' thus establishes the admission lap of the valve gear. An angle equal to the angle subtended by the cord *b* of Figure 24 is laid off at the center of the circle and bisected by the line 166 so that the cord B equals the cord *b*. This angle, indicated by the arc 175, is the angle of admission of air when operating at 600 pounds per square inch absolute. The center of the admission eccentric must therefore be shifted between the points 173 and 176 as the pressure changes from 150 pounds to 600 pounds. At intermediate pressures the center of the admission valve eccentric must be positioned at intermediate points on the line 174.

By way of example, a description will be given of the manner of locating the position of the admission eccentric at a pressure of 225 pounds per square inch in the high pressure main, it being understood that the same procedure is followed for other pressures. At 225 pounds pressure in the high pressure main the admission cut-off should take place at 54% of the stroke, as may be determined from the curve 50 of Figure 7. A point is located on the scale 166 of Figure 24 equal to 54% of the stroke and with that point as a center an arc is struck with a radius equal to the length of the connecting rod. The angular distance from the point 172 to the point of intersection of this arc with the circle 165 is then measured and half of it laid off of each side of the line 166. This establishes the position of the line 177. The point 177' of intersection of this line with the line 174 determines the requisite center of the admission eccentric in order to give cut-off at 54% of the stroke, which is the cut-off necessary when the pressure is 225 pounds per square inch absolute.

The determination of the exhaust lap and the point of closure of the exhaust valve and the compression cut-off for different operating pressures is determined in the same way as above described for the admission valve.

In order that a fuller understanding of the valve design may be had an example will be given of the valve design for adiabatic compression and adiabatic expansion between the limits of 100 pounds per square inch and 600 pounds per square inch absolute based upon an assumed clearance volume of 5% when the piston is at its innermost position within the cylinder. For this purpose reference may be had to the following table:

TABLE I

Adiabatic compression and adiabatic expansion between the limits of 100 p. s. i. and 600 p. s. i. abs. based on a clearance volume of 5%

1	2	3	4	5	6	7	8
Per cent Torque	Corresponding M. E. P. in p. s. i.	Corresponding high pressure, p. s. i. abs.	Admiss'n per cent Cutoff	Values in Col. #4 plus 5	5×105 divided by Values in Col. #5	Values in Col. #6 minus 5	Compression Cut-off equals 100 minus Values in Col. #7
465	203	600	24.2	29.2	18	13	87
405	177	475	29.4	34.4	15.3	10.3	89.7
352	154	400	34.1	39.1	13.4	8.4	91.6
306	134	333	39.0	44	11.9	6.9	93.1
266	116	288	44.2	49.2	10.7	5.7	94.3
232	101	252	49.5	54.5	9.6	4.6	95.4
201	88	225	54	59	8.9	3.9	96.1
175	76.5	207	59	64	8.2	3.2	96.8
152	66.5	185	62.9	67.9	7.7	2.7	97.3
132.2	57.7	170	67	72	7.3	2.3	97.7
115	50.2	160	70	75	7.0	2.0	98
100	43.6	150	73.5	78.5	6.7	1.7	98.3

The figures in this table are arrived at as follows: The maximum pressure in the high pressure main is set at 600 pounds. This figure is placed at the top of column 3. Planimeter readings taken from the curves of Figure 7 show that with a high pressure of 600 pounds absolute per square inch and a low pressure of 100 pounds absolute per square inch the corresponding mean effective pressure is 203 pounds per square inch above the low pressure. This determines the top figure in column 2. Assume that at the lowermost operative pressure the torque is to be taken at unity, or 100%. This is the figure placed at the bottom of column 1. Assume that it is desired to have a torque control range in twelve steps, each step to be 115% of the next lower value. The values in column 1 are then filled in starting with the lowermost value of 100% and working upwardly so that each higher value is 115% of its preceding lower value. This gives the values of column 1 and gives a maximum torque of 465% of the minimum torque. Since the mean effective pressures are proportionate to the torques, the corresponding values in column 2 may be obtained, being directly proportionate to the corresponding values in column 1. The values in column 2 are then ascertained since the top value of 203 is known. This gives the mean effective pressures required to obtain torques corresponding to the values of column 1. The values in column 3 are values of the pressure required in the high pressure main to give a mean effective pressure of the corresponding value in column 2 as determined by planimeter readings. That is, the indicator diagram must reach high pressures such as indicated in column 3 in order that planimeter readings shall give corresponding mean effective pressures as set forth in column 2. Column 4 gives the requisite admission cut-off for the corresponding pressures of column 3, said admission cut-offs being in terms of percent of the piston stroke and being obtained from the curve **50** of Figure 7. Column 5 is the same as column 4 with 5% added thereto for the 5% clearance. Columns 6 and 7 are intermediate values that are used for obtaining the values in column 8. An explanation will first be given of how the values in column 6 are obtained, for which reference should be had to Figure 7. At 600 pounds pressure, the curve **50** shows that the admission cut-off should be at 24.2% of the stroke or at a value of five points more than that when measured not from the base line 52, which is the end of the stroke, but from a line 53, which represents the clearance line and therefore is 5% to the left of the base line **52**. It is also known that the curve **51**, which is an adiabatic compression curve, must reach 600 pounds pressure absolute at the end of the stroke or at zero stroke. It is now required to find the point **54** at which the curve **51** must intersect the abscissa. Since the curves **50** and **51** are both adiabatic curves the distance L_1 is to L_2 as the distance X, which is the distance of the point 54 from the base line **53**, is to 105, which is the distance from the base line **53** to the point where the curve 50 intersects the abscissa. Setting up an equation we now have

$$\frac{L_1}{L_2}=\frac{X}{105}$$

L_1 is known as 5. Therefore

$$X=5\frac{(105)}{L_2}$$

For any given pressure the value of L_2 may be read from Figure 7, the value being five greater than the abscissa of the curve 50 at the corresponding pressure. The value of L_2 for different pressures are therefore the values given in column 5 of Table I. Thus for any given operating high pressure the corresponding compression curve **51** must intersect the base line at a distance from the line **53** given by the equation above. This gives the values for column 6. The values in column 7 are just five less than the values in column 6 and therefore give the distance from the end of the stroke at which the exhaust valve must be closed in order that upon completion of the stroke the air which was in the cylinder at the time of closure of the exhaust valve will be compressed to the desired pressure, as set forth in column 3. Column 8 is merely 100% minus the value in column 7, which indicates that compression commences upon the completion of that portion of the stroke of the piston.

It is to be noted that the action of the two automatic valves of each cylinder is controlled entirely by the difference between the pressure in the engine cylinder and the pressure in the high and low pressure mains, as the case may be, and that the opening and closing of the automatic valves is independent of the action of the pressure controlled cut-off. Therefore the automatic valves operate in the same manner regardless of the point of the admission cut-off or exhaust cut-off. If the automatic pressure responsive mechanism were to become disabled it could be locked to operate the admission cut-off at any appropriate fixed position without in any way interfering with the action of the two automatic valves of each cylinder. Since the two mechanical valves and the two automatic valves do operate effectively regardless of the pressure at which the mechanical valves are operated, it follows that the automatic valves can be used in combination with the mechanical valves of any air engine and will be valuable on an air engine that operates at a fixed cut-off.

An explanation will now be given of the construction of the governor and reversing device, for which reference should be had to Figures 8, 21 and 22. The function of the governor is to so time the operation of the mechanically operated valves of the engine that while the engine is operating on a load the valves will open on constant lead and will close slightly in advance of the theoretically correct cut-off so that in the next valve operation the automatically operated valves will open slightly ahead of them. In the embodiment of the invention here illustrated the governor is designed for stationary mounting, its movement being transmitted to the shaft **152** and through that shaft and differential to the cams. The primary force that reverses the engine and actuates the governor is the air pressure in the high pressure main. The governing and reversing mechanism **22** includes a stationary double acting governing cylinder **180**, a movable double acting reversing cylinder **181**, a long partially hollow piston rod **182** interconnecting the two cylinders, pistons **183**—**184** on the piston rod **182**, and a governing spring **185** acting on the piston rod **182** to urge it to a central position. The piston rod **182** has collars **186**—**187** keyed thereon. Between these collars there is located tubes **188** and **189** through which the piston rod **182** extends. The tubes **188** and **189** have annular plate-like flanges **190**—**191** against which the opposite end of the spring **185** bears. The flanges **190**—**191** and the spring **185** are located within a casing **193** mounted in any desired manner upon the governing cylinder **180**.

One end of the piston rod 182 has two incommunicable ducts 196—196' extending lengthwise therethrough. The duct 196 interconnects one side of the govening cylinder 180 with a corresponding side of the reversing cylinder 181. The other duct 196' connects the other sides of the same two cylinders together. The reversing cylinder 181 has a gear rack 194 secured thereto which is in mesh with a pinion 195 that is keyed to the shaft 152 of the differential. Air under pressure from the high pressure main 8 is supplied under the action of the controller 15, as will be more fully described as this description proceeds, to one or the other of the tubes 23—24 and to one side or the other of the governing cylinder 180. The pressure on one side of each piston 183—184 is thus the pressure in the high pressure main 8 whereas the pressure on the other side of each piston 183—184 is atmospheric pressure.

Figures 8 and 21 show the reversing cylinder 181 in the position that it occupies when pressure is applied to the pipe 24. At that time air at high pressure enters the cylinder 180 on the left hand side of the piston 183 and tends to move the piston 183 and the piston rod in a direction to the right as seen in Figure 21. The piston rod moves to the right and through the action of the collar 188 and the tube 188 compresses the spring 185. The piston 183 continues to move to the right until the pressure of the air admitted into the cylinder 180 through the pipe 24 is exactly counterbalanced by the spring 185. During the movement of the piston 183 to the right the piston 184 which is connected therewith moves the floating cylinder 181 with it. This cylinder and piston 184 move to the right as one unit, since the air duct 196 maintains the pressure in the cylinder 181 on the left hand side of the piston 184 the same as that in the high pressure main, whereas the duct 196' maintains the pressure on the right hand side of the pistons 183—184 the same as in the pipe 23, which is maintained at atmospheric pressure through the controller, as will be more fully set forth as this description proceeds. Thus any change in pressure in the high pressure main results in a corresponding change in the pressure in the pipe 24 and a corresponding movement of the piston rod 182 against the action of the spring 185. The moving piston rod 182 carries with it the reversing cylinder 181 which actuates the gear 195 to actuate the differential 150, in the manner previously described, to turn the cam shaft actuating the cams 125—126. The angular position of the cams with respect to the crank shaft 33 is thus determined by the amount of movement of the cylinder 181 which in turn is determined by the pressure in the high pressure main and the action of the spring 185. For any given pressure in the main 24 there is a corresponding angular position of the cams 125—126 with respect to the crank shaft 3. The movement of the governor piston 183 compresses the spring 185 and is thereby limited to that point at which the pressure on the piston 183 is balanced by the pressure of the spring 185. Thus the position of the cams for one direction of rotation of the air engine is determined by the action of the governor.

For the opposite direction of rotation of the air engine the positioning of the cams is also determined by the action of the governor. To obtain a reverse direction of rotation from that previously described, the controller 15, to be presently described, is set so as to apply atmospheric pressure to the pipe 24 and to apply the pressure of

the high pressure main 8 to the pipe 23. High pressure from the pipe 23 is immediately transmitted through the duct 196' in the piston rod to the portion of the cylinder at the right hand face of the piston 184 and the pressure in the portion of the cylinder 181 on the opposite face of the piston 184 is made atmospheric pressure by the communicating duct 196 in the piston rod. The cylinder 181 therefore immediately moves a full stroke to the right, thus immediately actuating the gear 195 to turn the differential and thereby turn the eccentric cams 125—126 to set the engine valves for reverse direction of operation. The piston 183 moves to the left under the action of pressure supplied thereto from the pipe 23, thereby causing the collar 187 to force the tube 189 and plate 191 to compress the spring 185 so that the ultimate position of the piston 183 in its movement to left from the position illustrated in Figures 20 and 21 is again that position at which the air pressure on the piston 183 is exactly counterbalanced by the spring 185. Any change in pressure in the high pressure main results in a change in position of the piston 183 and corresponding shifting of the cylinder 181 and turning of the gear 195 to reset the eccentric cams 125—126 to a new position which is a function of the pressure in the high pressure main.

The ultimate positions of the eccentric rings with respect to the crank shaft for any given pressure value in the high pressure main are determined by the shapes of the cams 125—126 and the angular movement of the cams as determined by the spring 185 and the pressure in the high pressure main. The shapes of the cams and the calibration of the spring 185 must therefore be correlated. In the engine here described they are so correlated that an indicator diagram taken from any one of the engine cylinders will follow the adiabatic curves 50—51 to the particular pressure prevailing in the high pressure main. As the engine torque is increased, increments of additional air pressure must be increased at a faster rate than the desired increase of increments of torque. Compensation for this disproportion can be made either by taking care of this factor in the cam design or by making the governor spring 185 so proportioned or controlled that it will develop progressively increased increments of counter pressure at the rate required to supply the proper degree of compensation. One such design of spring is illustrated in Figure 21. It is made up of wire or rod 201 of rectangular cross section formed into a helix of cylindrical shape. Nested with the spring is a helical spaced member 202 made up of thin rectangular lamination strips also formed into a helix. These strips are of various lengths so grouped that as the spring is compressed successive turns of the spring 201 will progressively close on the spacer member. By properly proportioning the lengths of the various laminations 202 there can be obtained any desired spring calibration within the limits of the design. This is merely one way of obtaining the desired calibration of the spring. An equivalent effect can be obtained by winding the turns of the spring 201 with different spacings.

The spring 185 in the governor is pre-compressed to a degree such that the governor is inoperative at air pressures under 150 pounds per square inch. At pressures up to 150 pounds per square inch absolute the spring 185 maintains the governor piston rod 182 in the position illustrated in Figure 21.

It is obvious from the above description that

the proper functioning of the engine is dependent upon proper functioning of the governor and that the spring design, correlated with the shape of the cam, constitutes the essence of the governing design.

An explanation will now be given of the construction of the controller 15 of Figure 1, which is shown more particularly in Figures 26 to 33 inclusive.

The functions of the controller are:

1. To set the reversing cylinder 181 of the governing and reversing mechanism 22 for the desired direction of rotation;

2. To operate the unloading devices on the air compressor in proper sequence and in such a manner as to obtain the desired torque and speed from the air engine;

3. To change the functioning of the engine from motoring to braking when desired;

4. To so control the flow of air through a by-pass between the high pressure main and the low pressure main as to obtain the desired braking torque and speed from the engine.

The control of all of these functions is centered in a single lever, the position of which determines the function to be performed and the values of speed and torque that the engine will develop. The controller 15 includes a base 250 upon which the various parts are mounted to constitute a unitary assembly. A pair of brackets 251—251 (Figs. 26, 27 and 30) carry a shaft 252 that is journalled in bearings in the brackets so that it is rotatable and also longitudinally movable. A hand operating lever 253 is keyed to the shaft and extends upwardly therefrom through a slotted arcuate plate 254 that is bolted or otherwise secured in position to and extends between the two brackets 251—251. The plate 254 is of a shape such as is shown more particularly in Figures 30 and 32 and has a U-shaped slot therein consisting of two parallel longitudinal slots 255—256 joined by a cross slot 257. The movement of the lever 253 is guided by the slots 255, 256 and 257. The lever 253 may be moved in either of the slots 255—256 to produce a corresponding oscillation of the shaft 252, or it may be moved from one of the slots to the other, through the cross slot 257, producing a corresponding longitudinal or axial movement of the shaft 252. A gear 260 is keyed to the shaft 252 and is in mesh with and drives an elongated pinion 261 keyed on a rotatable cam shaft 262, journalled in brackets 263—263 mounted on the base 250. The shaft 252 is mechanically connected to a piston rod 264 that carries two pistons 265—266 in a cylinder 267, of a direction controlling pilot valve 268, as may be seen from Figure 27. A tube or conduit 269 connects the cylinder 267 to the high pressure main between the pistons 265—266. Tubes 23 and 24 on opposite sides of the tube 269 extend, respectively, to the forward and reverse cylinder outlets of the reversing device 22, in accordance with the connections illustrated in Figures 1 and 21. When the pistons 265—266 are in the positions illustrated in Figure 27 communication is established from the high pressure main by way of the tube 269 to the reverse direction controlling tube 24. At the same time the forward direction controlling tube 23 is open to atmosphere at the cylinder 267. If the piston rod 264 is moved to the right from the position illustrated in Figure 27 the tube 24 will be opened to atmosphere at the cylinder 267 and the tube 23 will in turn be connected to the high pressure main via the tube 269. Movement of the piston rod 264 to the right from the position illustrated in Figure 27 is effected by manually shifting the hand operated lever 253 crosswise through the cross slot 257, thus moving the shaft 252 to the right and with it the pistons 265—266.

The cam shaft 262 has keyed thereto a plurality of cams, in this instance six in number, indicated at 271 to 276 inclusive, so that rotation of the cam shaft results in rotation of all six of the cams. The cam 276 is illustrated in Figure 31. This cam controls a bell crank lever 278 pivoted at 279 and carrying at one end a cam roller 280 in engagement with the cam 276, and at its opposite end through a link 281 controlling the position of a piston 282 in the cylinder of a pilot valve 283, which piston in one position, that illustrated in Figure 31, establishes communication between a manifold 284 that is connected to the low pressure main, and a flexible metal tube 16 that extends to the unloader of the inlet valve of one of the cylinders 2 of the compressor. A manually operable three-way valve 285 is interposed between the tube 16' and the cylinder 283 for a purpose to be more fully set forth as this description proceeds. At the present it is sufficient merely to point out that a valve 285 is provided in only one of the six unloader controlling tubes 16 leading from the controller 15 to the unloaders on the cylinders 2, as may be seen from Figures 1 and 26. The piston 282 in its alternate position in the cylinder 283 (Fig. 31) closes off communication between the tube 16 and the low pressure manifold 284 and opens the tube 16 to atmospheric pressure, thereby disabling the corresponding unloader and allowing the corresponding compressor valve controlled by the tube 16 to operate in its normal manner. Each one of the cams 271–276 operates a similar bell crank 278 to actuate a pilot valve 283 to control different unloaders through pressure applied from the low pressure main to the corresponding tube 16. The cams 271–276 are so arranged as to give the proper sequence of opening and closing of the unloader valves of the compressor cylinders 2. When the controller handle 253 is in the neutral position illustrated in Figure 27, all of the cams 271–276 control their respective pilot valves 283 to apply pressure to the respective unloading devices so that the respective intake valves of the compressor cylinders 2 are all held open. The handle 253 has fourteen operative positions to one side of its neutral position in each of the slots 255—256.

A sequence diagram for the operation of the cams 271–276 is illustrated in Figure 33. In position 1 cam 276 releases its unloading device. In positions 2, 3 and 4 cams 275, 274 and 273 successively release their unloading devices. In position 5 cam 272 releases its unloading device and cams 273 to 276 again operate their unloading devices. In positions 6, 7, 8 and 9, respectively, cams 276 to 273, respectively, successively release their unloading devices. In position 10 cam 271 releases its unloading device and cams 273 to 276 actuate their unloading devices. In positions 11 to 14 inclusive, cams 275 to 273 successively release their unloading devices. The full lines 290 in Figure 33 indicate the positions at which the unloading devices controlled by the respective cams are released, which correspond to the positions of the controller during which the corresponding air compressor cylinders of the air compressor are functioning.

As previously stated, the air driven compressor unit 1 consists of eight separate cylinders 2.

Four of the cylinders are shown in Figure 1, the other four being located immediately below the four cylinders shown in Figure 1. In the preferred arrangement four of the cylinders are of uniform size and are small cylinders. The other four cylinders are of uniform size and are large cylinders. Each of the large cylinders is preferably of a diameter which is 1.58 times the diameter of a small cylinder or a cross sectional area 2.5 times the cross sectional area of a small cylinder. Cams 273, 274, 275 and 276 each control the unloader of one small cylinder. Cam 271 controls the unloaders of two large cylinders simultaneously. Cam 272 controls the unloaders of the remaining two large cylinders simultaneously. If the output of a small cylinder, per revolution of the engine drive shaft 6, is taken as unity, then the outputs of each large compressor cylinder per revolution of the engine drive shaft 6 is 2½, and the output for each pair of large cylinders, per revolution of the shaft 6, is 5.

The controller 15 also includes a brake control cylinder 290 that has pistons 291—291′ therein that control the establishment of communication between the high pressure main and the low pressure main through conduits 292—293 that lead respectively to the high pressure main and the low pressure main. The pistons 291—291′ are connected to a piston rod 294. The piston rod 294 is connected through a link 295 and pin 236 to a pair of spaced parallel bell crank levers 297 that are pivoted at 298 to the brackets 251 and carry at their opposite end a roller 299 that is engaged by the hand operated lever 253. If the lever is moved back from the neutral position of Figure 30. As that lever 253 is moved back from the neutral position, that is, to the left from the position illustrated in Figure 30, it engages the roller 299 and swings the bell crank 297 counterclockwise to force the piston 291 progressively downward to progressively uncover more and more of the port areas in the cylinder 290 communicating with the conduit 293, thus progressively establishing a greater and greater flow of air from the high pressure main to the low pressure main.

An explanation of the operation of the system thus far described will now be given. Assume that the engine driving the compressors 2 is driven at a constant speed. Assume that the controller is in its neutral position. All of the unloader devices of the six compressors 2 are energized and therefore hold the automatic inlet valves of the compressor continuously open. Therefore no air is being forced from the low pressure main to the high pressure main. The auxiliary compressor is maintaining the pressure in the low pressure main constant at 100 pounds per square inch absolute. The controller handle is then moved to its first position. This releases the pressure on the unloading device of one of the compressor cylinders 2. That cylinder will commence to move air from the low pressure main to the high pressure main, delivering the air to the high pressure main at a constant rate. The pressure in the high pressure main will therefore build up. The compressed air engine 20 commences to operate. It will operate at a speed determined by the torque of its load and the pressure in the high pressure main. For each revolution of the compressed air engine a fixed quantity of air will be withdrawn from the high pressure main and returned to the low pressure main. As long as the rate of rotation of the engine 20 is such that the rate of withdrawal of air from the high pressure main by the engine is less than the rate of delivery of air thereto by the compressor, the pressure will continue to rise in the high pressure main. As the pressure continues to rise the engine 20 accelerates until ultimately it reaches a speed at which the rate of air withdrawal from the high pressure main by the engine 20 exactly equals the rate of input of air into the high pressure main by the compressor. The engine 20 will then continue to operate at that speed. Air thus circulates in a closed system from the low pressure main to the compressor, thence to the high pressure main, then to the compressed air engine and back to the low pressure main. At that time the auxiliary air compressor 10 merely serves to supply the low pressure main with an amount of air necessary to compensate for leakage. Should it be desired to increase the speed of the engine 20 it is merely necessary to increase the rate of air delivery to the high pressure main by the compressor. This is done by moving the controller to its second position thereby releasing the unloader device for another small compressor cylinder. The rate of air delivery into the high pressure main is thus doubled, and the pressure commences to rise. This causes the compressed air engine to accelerate thereby increasing the rate of withdrawal of air from the high pressure main. When the engine reaches such a speed that its rate of air withdrawal from the high pressure main again equals the rate at which air is delivered to the high pressure main by the compressor, equilibrium is established, and the engine will continue to operate at its new constant speed. To further increase the speed of operation of the engine it is necessary to release another unloader valve to bring another small compressor cylinder into operation. The controller in positions 1, 2, 3 and 4 brings into operation first one, then two, then three, and then four of the small cylinders. In position 5 the controller releases the unloading devices of two large cylinders and operates the unloading devices of the four small cylinders. The two large cylinders have a combined air output of five times the output of one small cylinder.

As previously stated, the speed of operation of the engine is determined by the rate of air delivered by the compressor into the high pressure main. The pressure in the high pressure main, at which the compressed air engine is operating, is determined by the torque of the load. It is desired that the pressure in the high pressure main should not rise above 600 pounds per square inch absolute. If the torque of the load is such as to require more than 600 pounds per square inch in order to move it, the load will not move, and as the pressure in the high pressure main rises above the 600 pound value due to the continued delivery of air to the high pressure main by the compressor without a corresopnding withdrawal of air by the engine, the safety valve 13 will operate to by-pass air from the high pressure main directly to the low pressure main. If the compressor is driven by a constant speed internal combustion engine having a definite maximum horse power, that will limit the maximum horse power of the compressed air engine. Since the number of compressor cylinders in service determines the speed of the compressed air engine 20, it follows that with a constant horse power driving engine for the compressors 2 the maximum torque that can be developed by the engine 20 will vary inversely with its speed and therefore inversely with the number of compressor cylinders 2 that are in ser-

vice. The maximum possible torque for the compressed air engine will therefore increase as the number of compressor cylinders in service is decreased. At 600 pounds high pressure and 100 pounds low pressure the mean effective pressure is 203 pounds which develops the maximum torque possible for the engine **20**. If the torque of the load requires a mean effective pressure over 203 pounds (above the 100 pounds absolute pressure of the low pressure main) the load will not start, and the pressure in the high pressure main will tend to rise, but such rise is prevented by the safety valve. The arrangement is such that at a speed of the compressed air engine **20** equal to that attained when three small compressor cylinders are in service and at a maximum torque obtainable with a high pressure of 600 pounds per square inch, the horse power output of the compressed air engine **20** is equal to the maximum horse power output of the internal combustion engine driving the compressors **2**. This means that at the two lower speeds of the compressed air engine **20** obtained when one or two small compressor cylinders are in service, no larger torque can be carried by the compressed air engine and the internal combustion engine that drives the compressors therefore operates at below its maximum horse power.

From the above description it is apparent that the engine **20** permits a rise in pressure in the high pressure main to such a value and operates at such a speed that the torque developed by the engine exactly balances the torque of the load. If the torque of the load goes down (as in the case of a locomotive arriving at a decline in the road) the engine will tend to accelerate only momentarily, thereby withdrawing an additional amount of air from the high pressure main and thus reducing the pressure therein and eliminating the tendency to accelerate.

In the system of Figure 1 the compressed air engine **20** may also be used as a brake for the load. Assume that a load, such as a train of railroad cars, is driven by the engine **20** and it is desired to brake the speed of the train. The controller handle **253** is moved to its extreme back position, at the cross slot **257**. The unloaders of the compressors **2** are energized, thereby stopping the delivery of air to the high pressure main. The pressure in the high pressure main immediately drops due to the withdrawal of air therefrom by the engine **20** and due to the establishment of a by-pass from the high pressure main to the low pressure main at the brake control cylinder **290** as the handle **253** is moved to the left of the position illustrated in Figure 30. The controller handle **253** while in this position is shifted through the cross slot **257** thereby actuating the pistons in the reversing pilot cylinder **268**. This disconnects the high pressure pipe 233 (Fig. 27) from the tube **24** and connects it to the tube **23**. The tubes **23** and **24** lead to the governing and reversing device, as previously explained. By reference to Figure 21 it may be seen that the changing of the application of air prressure from the tube **24** to the tube **23** results in the application of pressure by way of the tube 23, to the part of the cylinder on the right hand side of the piston **183** and the passageway **196'** to the part of the reversing cylinder on the right hand side of the piston **184**, thereby moving the reversing cylinder **181** to the right from the position illustrated in Figures 20 and 21 to the position illustrated in Figure 22, thus actuating the gear 193 and the differential **150** to set the cams **125** and **126** of the eccentrics that control the mechanical valves of the engine **20** for reverse direction of operation. The locomotive now drives the engine **20** as a compressor taking air from the low pressure main and delivering it to the high pressure main. At this time the automatic valves **58** and **58'** of the engine of Figure 2 will open before the mechanically operated valves open, and close after the mechanically operated valves close. The automatic valves therefore become primary valves while the mechanically operated valves serve no useful function.

As the engine **20**, now operating as a compressor, delivers air into the high pressure main the pressure in that main tends to rise. A rise in pressure is, however, prevented due to the discharge of air from the high pressure main to the low pressure main through the brake control cylinder **290** because the brake controller handle **253** is in its extreme left hand position. If the handle is moved to the position illustrated in Figure 30, thereby completely shutting off the by-pass between the two mains at the braking cylinder **290**, the pressure will rise to its maximum value. As the engine continues to move the additional air forced thereby into the high pressure main is discharged into the low pressure main through the safety valve **13**. The pressure therefore remains at 600 pounds and the locomotive receives the maximum braking effect until it comes to rest. If it is desired to reduce the braking effect it is merely necessary to move the controller back to the left of the position illustrated in Figure 30. As the hand operated lever **253** is moved back it acts on the roller **299** and bell crank **297** to force the piston rod **294** of the brake control piston **293** to move downwardly and thus establish an additional by-pass from the tube **292** that is connected to the high pressure main to the tube **293** that is connected to the low pressure main. This reduces the pressure in the high pressure main thereby reducing the braking effort. The amount of reduction of the braking effort is controlled by the amount of downward movement of the braking piston **291** by leftward movement of the handle **253**. As the locomotive speed decreases, the rate at which air is forced into the high pressure main by the engine **20** operating as a compressor also decreases, and when the handle **253** maintains the braking control piston **291** uncovering the ports leading to the low pressure conduit **293** the pressure in the high pressure main gradually drops to the pressure in the low pressure main, thus gradually reducing the braking effect.

An explanation will now be given of the application of the power system of Figure 1 to the draw works of an oil well drilling rig, for which reference may be had to Figure 34. The air compressor unit **1** of Figure 1 is shown as located near the usual type of drilling rig **310**. The drill pipe to the bottom of which the drill is connected is indicated at **311**, said pipe being arranged to be rotated in the usual manner through a gear mechanism within a box **312** driven in any desired manner, as by a belt **313** leading to an engine. The engine may, optionally, be the same engine that drives the compressor **2**, the belt **313** being shiftable from an idler pulley to a driving pulley on the shaft of that engine to start the drilling. A connection is provided at **315** for circulating mud through the bore to carry away the rock and other material loosened by the drill, all in a manner known in the art

The weight of the drill pipe 311 is supported in a novel manner in order to permit the air engine to carry any desired fractional part of the weight, thereby controlling the remaining weight or pressure exerted by the drill bit at the bottom of the bore. To that effect a weighing device 320, shown more particularly in Figure 35, is interposed between the drill pipe 311 and the travelling block 321 that supports the drill pipe. The weighing device consists of a plate 322 which is suspended at 323 from the pulley block 321. The plate 322 has a pair of links 325—325 suspended therefrom, which links in turn support a bowed leaf spring assembly 326. A yoke 327 is supported at the center of the spring 326 and in turn supports the drill pipe 311. The pull of the drill pipe on the plate 322 determines the amount of deflection of the spring 326. The deflection of the spring moves a pointer 328 pivoted at 329 to the yoke 327 and at 330 to an extension of the plate 322. The pointer 328 moves over a calibrated scale 331 so that the position of the pointer on the scale 331 indicates the amount of downward pull of the drill pipe 311 on the plate 322 or, conversely, the upward pull of the plate 322 on the drill pipe through the pulley arrangement. The cable 335 of the block and tackle, which includes the block 321, extends to a reel or drum 336 driven by the air engine 20 of Figure 1 that receives its air from the compressor unit 1 of Figure 1, through a controller such as shown at 15 of Figure 1. The drum 336 is driven by the engine to raise or lower the pipe 311 in the manner known in the art. The speed of the drum is controlled by the controller 15 in the manner previously described. During the actual drilling operations there is no hoisting.

If an unloader of one of the small compressor cylinders 2 is released that compressor will start to build up a pressure in the high pressure main so that the pistons of the engine 20 will apply a torque tending to turn the drum 336 and raise the block 321. As this torque increases it progressively exerts a greater and greater upward pull on the pipe 311 through the plate 322, thus progressively reducing the weight or pressure of the bottom of the drill bit in the bore drilled thereby. Since the total length of the drill pipe 311 is known the weight thereof is also known. The pressure of the drill is, therefore, the difference between the known weight of the drill pipe and the upward pull exerted thereon by the air engine, as indicated by the pointer 328 on the scale 331. If it is desired to maintain constant the weight of the drill bit in the bore it is merely necessary to maintain the pointer 328 in a constant position over the scale 331, this being maintained by maintaining a constant pull on the cable 335 by the air engine. The constant pull is maintained by maintaining a constant pressure in the high pressure main. When the pressure in the high pressure main reaches the desired constant value, as indicated by the pointer 328 reaching its desired position on the scale 331, the unloader of the compressor cylinder that was operating must be actuated, to stop a further rise in pressure in the high pressure main. A relay is provided for automatically accomplishing this result, and for recommencing the operation of the compressor if the pressure falls below the set value.

The constant pressure regulating relay is shown in Figures 38 and 39, the connections of that relay being shown in Figure 1. This relay is connected to the first tube 16 at the controller by turning the three-way valve 285 through 90° in a clockwise direction from the position illustrated in Figures 1 and 26. The relay consists of a cylinder 340 which is connected to the high pressure main and balances the pressure of the high pressure main on one side of the piston 341 against a compression spring 342 whose tension is adjustable by a cap screw 343 threaded into the cylinder 340. The piston 341 actuates a piston rod 344 that carries three pistons 345, 346 and 347 in a cylinder 348. The space in the cylinder 348 between the pistons 346 and 347 receives air from the low pressure main via a tube 350 that is adapted to discharge through a tube 351 under control of the piston 347. If the pressure in the high pressure main becomes too low, as due to leakage of air, the spring 342 forces the piston 341 upwardly thereby opening the tube 351 to atmosphere at the cylinder 348. This atmospheric pressure is conveyed via the tube 351, the three-way valve 285 (Figs. 1 and 26) and the tube 16 that leads to the unloader valve of one of the small compressor cylinders. This releases the unloader valve of that particular engine cylinder by removing the pressure from that unloader. That cylinder therefore commences to force air into the high pressure main to build up the pressure therein. As the pressure builds up, the piston 341 moves downwardly. When the pressure reaches the desired value the piston 341 has moved downwardly an amount just sufficient to apply low pressure from the low pressure line 350 through the cylinder 348 to the tube 351, thence through the valve 285 to the right hand end line 16 of Figures 1 and 26, thereby energizing the particular unloader valve and stopping a further building up of the pressure in the high pressure main. Should the pressure in the high pressure main become excessive then the piston 341 will move downwardly against the action of the spring 342 with the result that the piston 346 will uncover the opening to a line 355 which establishes communication between a line 356 leading from the high pressure main and a line 355 that leads to the low pressure main. This will bleed off the excess pressure from the high pressure main. Thus the relay of Figure 38 will constantly regulate to maintain a pressure in the high pressure main necessary to maintain the piston 341 balanced against the spring 342 as set by the screw 343. The pressure of the drill bit in the bore is thus maintained constant.

As the length of pipe line 311 is increased it is necessary to increase the pressure in the high pressure main in order that the remaining pressure of the drill bit shall remain constant. To increase the pressure in the high pressure main it is merely necessary to adjust the screw 343 to increase the compression of the spring 342. This momentarily causes the piston 341 to rise and, through the piston rod 344, to close off the application of pressure from the low pressure main 350 to the tube 351 and expose that tube to atomspheric pressure thus releasing the unloader of the compressor controlled thereby so that the pressure commences to build up and is then maintained at a new value required to maintain the piston 341 in the position illustrated in Figure 38, when balanced against the higher spring pressure.

When it is desired to raise the pipe 311 it is necessary to slip the belt 313 to the idler pulley on the shaft and thus discontinue the rotation of the pipe 311. Thereafter the hand operated valve controlling the application of pressure to the relay 33 is closed and the three-way valve

285 is set to the position illustrated in Figure 26. This closes off the connection between the first tube **16** and the tube **351** leading to the constant pressure relay and instead connects the first tube **16** to the controller **15** so that the unloader of the particular cylinder connected to the first tube **16** is now controlled by the controller **15**. The controller may then be used in a manner previously described to hoist the load at the desired speed.

When the hand lever is at the neutral position or to the left thereof none of the compressor cylinders **2—2** are delivering air to the high pressure main because all of the compressor unloaders are actuated. Also, when the lever is to the left of the neutral position the brake control cylinder permits air flow from the high pressure main to the low pressure main. This mode of operation is utilized for braking.

When it is desired to lower the pipe line the air engine **20** may be used as a brake to brake the descent of the line. This is accomplished in the following manner: The hand operated lever **253** is shifted in the cross slot **257** to set the engine valves for the corresponding direction of rotation. Shifting of the hand lever **253** in the slot **257** is of no effect on the operation of the unloaders because the gear sector **260** merely slides lengthwise on the pinion **261**. This shifting of the lever, however, shifts the pistons **265—266** of the direction pilot valve **268** to the right from the position illustrated in Figure 27 and applies the pressure from the high pressure main to the tube **23**, thus positioning the reversing cylinder **181** of Figure 20 in the manner previously described, to set the eccentrics that control the mechanical valves of the engine **20** for the proper direction of rotation. The load now commences to drive the engine **20** as a compressor. The controller handle is shifted to the position illustrated in Figure 30. The engine **20** builds up pressure in the high pressure main until the pressure in the high pressure main becomes sufficient to counterbalance the torque of the load, at which time the load comes to rest. In the position of the controller handle illustrated in Figure 30, the maximum braking effect is obtained. If it is desired to obtain a smaller amount of braking effect the handle **253** is moved back, that is, to the left from the position illustrated in Figure 30. This immediately causes the piston **291** to move downwardly and establish communication between the lines **292** which is connected to the high pressure main and the line **293** which is connected to the low pressure main. Air immediately commences to flow from the high pressure main to the low pressure main, thus tending to relieve the pressure in the high pressure main and permitting a further descent of the load. The rate of air flow from the high pressure air main to the low pressure air main will determine the rate at which the compressed air engine **20**, now operating as a compressor, will rotate, and thus will determine the rate of descent of the load. As the hand lever **253** is moved further and further to the left, from the position illustrated in Figure 30, it uncovers progressively larger port areas in the cylinder **290** communicating with the line **293**, thus progressively increasing the rate of air transfer from the high pressure main to the low pressure main and thereby permitting greater descending speeds of the load or, stated in other words, reducing the braking effect.

Movement of the controller to the right of the braking position, either in the slot **255** or the slot **256**, produces a progressively faster driving effect. Thus during hoisting operations, if the controller handle is moved from the hoisting position to the position illustrated, the load will come to rest due to the gravitational decelerating effect and then, without shifing of the lever in the slot **257**, it would tend to reverse its direction of rotation. As soon as the direction of rotation changes the hoisting engine commences to operate as a compressor, builds up pressure in the high pressure main and brings the load to rest when the pressure in the high pressure main has reached a value equal to the torque of the load. It is possible to control the deceleration of the load at will by controlling the lever **253**, either by moving the lever toward the slot **257**, or across the slot to set the engine valve cams for reverse direction, or in one or the other of the two slots **255—256** to cause the engine driven compressor to build up pressure in the high pressure main to drive the load in the opposite direction.

In the case of a load, such as a train of cars running on a level track, the retarding force is not a constant like the force of gravity but decreases as the speed decreases until it finally reaches zero at zero train speed and then does not reverse its direction of travel as in the case of a weight being hoisted. Because of these conditions, the braking action in the case of a locomotive is obtained by reversing the valve gear of the air engine, thereby changing the functions of the air engine to an air compressor. In the case of a hoisting load the mere change of the direction of travel of the load, due to the action of gravity, causes the engine **20** to commence to operate as a compressor.

The principles of the present invention can be applied to a power system which affords regenerative braking. Such a system is illustrated in Figure 40. In this figure a low pressure main is indicated at **360**. The pressure in this main is maintained at a constant value of, say, 100 pounds per square inch absolute by an auxiliary compressor **361**. A constant speed electric motor **362** drives a compressor unit **363** that receives air from the low pressure main **360** and discharges into a high pressure main **364**. A power unit **20**, of the same construction as the unit **20** of Figure 1, receives air from the high pressure main and discharges into the low pressure main, said unit **20** operating in the same manner as does the unit **20** of Figure 1, and including a governing and reversing device **22** which receives air from the high pressure main through a direction controlling pilot valve **268**, of the same construction as the pilot valve **268** of Figure 27. The shaft **252** of the direction controlling pilot valve **268** is manually operated by a hand lever **366** independent of the controller handle. The engine unit **363** is of a construction substantially identical with that of the unit **20**, differing therefrom only in that pneumatically controlled unloader valves and pneumatically controlled auxiliary valves are provided for a purpose to be more fully set forth as this description proceeds. It is sufficient here to state that when the load is being driven the unit **363** operates as a compressor driven by the electric motor **362** to supply pressure to the high pressure main **364** and that during regenerative braking the unit **363** acts as a motor, driven by pressure in the high pressure

main to drive the dynamo electric machine 382 as a generator to supply electric energy to the electric line connected to the machine 382.

The unit 363 always rotates in the same direction but is provided with a valve gear reversing device 22 of exactly the same construction as that of the unit 20 which, when set in one direction, causes the unit 363 to operate as a motor, and when set in the other direction causes it to operate as a compressor. The governing and reversing device 22 is controlled by a direction controlling pilot valve 268' like the direction controlling pilot valve 268 of Figure 27. The direction controlling piston rod shaft 252 of this pilot valve is controlled by a hand operated lever 366'.

The system of Figure 40 includes also a controller 15' similar to the controller 15 of Figure 27. The controller has a neutral position from which it is movable in one direction to actuate the brake cylinder 290 in a manner illustrated in Figure 30, and is movable in the opposite direction to progressively release pressure from fourteen pilot valves 283 similar to the pilot valves 283 of the controller of Figure 26. When the pressure is applied to the pilot valves 283 it is transmitted through the respective tubes 16a to unloaders and pneumatic valves of the engine 363.

The pneumatically controlled valves of the engine 363 are illustrated in Figures 41, 42 and 43. In Figure 41 the high pressure header is indicated at 44 and the low pressure header at 46, said headers being connected respectively through pipes 43 and 45 to the high pressure valve head 41 and the low pressure valve head 42 of one of the engine cylinders in the same manner as in the engine 20, as illustrated in Figures 2, 5 and 6, except that a pneumatically operated valve 370 is interposed between the high pressure header 44 and the pipe 43. The valve 370, illustrated more fully in Figure 42, is maintained normally open by a compression spring 371 and is adapted to be moved to a closed position by the application of pressure to a cylinder 372, which pressure acts on a piston 373 to close the valve. Once the pressure is released in the cylinder 372 the valve automatically opens under action of the spring 371. Pressure is applied to the cylinder 372 by way of a tube 374 which connects to the tube 16a. When the pressure in the tube 16a that leads to the cylinder illustrated in Figures 41 and 42 is reduced to atmospheric pressure, the spring 371 opens the valve 370 and permits direct communication or air flow from the header 44 to the pipe 43.

Each working cylinder of this engine is provided with mechanically operated high pressure and low pressure valves and with automatically operated high pressure and low pressure valves the same as the engine of Figure 2, and operated in the same manner. In addition the automatically operated low pressure valve of the engine cylinder 31 is provided with an unloader 375 which may be of any desired construction as, for instance, one such as shown in Marks' Mechanical Engineers Handbook, third edition, page 1373. As previously stated, the automatic valve consists of one or more rings or discs seated by a light spring and arranged to open when the pressure in the cylinder 31 drops slightly below the pressure in the low pressure main or low pressure valve head. The unloading device is arranged to maintain the automatic valve open mechanically, and consists of a piston 378 pressed upwardly by a spring 377 to its inoperative position and moved downwardly by pressure as applied through a tube 379. When the pressure moves the piston downwardly against the action of the spring 377 the piston moves a prong or group of prongs into engagement with the spring seated automatic valve and holds it in its open position.

The tube 379 is connected to the tube 16a, as shown in Figure 41, that leads from the particular cylinder of the engine, to the controller 15'. Thus when the controller 15' is set so that atmospheric pressure is applied to a particular tube 16a the valve 370 of the corresponding engine cylinder is maintained open by its spring 371 and the unloader 375 on the low pressure side of that cylinder is maintained inoperative by spring 377. That cylinder of the engine then operates as a compressor cylinder. If it is desired to disable that cylinder the controller 15' is moved to a position to apply pressure to a corresponding tube 16a. This pressure automatically causes the valve 370 to close, thereby closing off communication between the corresponding cylinder 31 and the high pressure header 44. At the same time the unloader valve 375 holds open the automatic valve controlling communication between the low pressure main and the cylinder 31 so that the cylinder 31 merely idles.

A description will now be given of the mode of operation of the system of Figure 40. The electric motor 362 is operating at a constant speed. Assume that it is desired to operate the unit 20 as a motor. The controller handle 253 is moved to progressively release the pressure on the different pilot valves 283 thereby progressively releasing pressure from successive tubes 16a and releasing the unloaders 375 of the cylinders. This permits the automatic valves on the low pressure side of those cylinders to function. At the same time the release of pressure from the line 16a causes the valve 370 of the corresponding cylinder to open and remain open. The cylinder then operates as a compressor under the action of the automatic valves and takes air from the low pressure main 360 and delivers it to the high pressure main 364. At that time the opening and shutting of the mechanical valves of the unit 363 as controlled by the governor 22 is of no effect because the automatic valves of that unit open in advance of the mechanical valves. The unit 363 thus acts as a compressor delivering air to the high pressure main to operate the unit 20. It is thus apparent that during normal operation, while the engine 20 is driving the load, the operation is the same as that of the system of Figure 1, the controller 15' determining the number of cylinders of the unit 363 that are in service, thus determining the rate of air delivery to the high pressure main, which in turn determines the speed of operation of the engine 20. The hand lever 366 controls the reversing pilot valve 268 to determine the direction of rotation of the engine 20. Therefore it is not necessary to have the two slots 255—256 of Figure 32.

For non-regenerative braking action the controller is brought to the neutral position which will give the maximum braking effect and may then be moved backward from that position to control the braking cylinder 290 in the manner previously described, to produce a diminishing braking action. To effect this braking action the hand lever 353 may or may not be moved to reverse the valve gear of the engine 20, depending

upon the type of load involved, as in the system of Figure 1 as previously described.

An explanation will now be given of the manner of operation of the system of Figure 40 for regenerative braking. This braking may be of either of two types, namely, the regenerative braking necessary to prevent or control acceleration of a load as, for instance, a train of cars on a decline, or to effect a deceleration of a load as, for instance, a train of cars on level ground. During regenerative braking the unit 363 must operate in the same direction as before, but now it must drive the machine 362 as a generator instead of being driven by that machine as a motor. The unit 363 must therefore act as a motor during regenerative braking whereas previously it acted as a compressor. To permit the unit to act as a motor it is necessary to reverse the valve gear thereof. This is done by actuating the reversing pilot valve 268′ by the handle 366′. The unit 363 may now act as a motor. The reversing valve handle 366 that controls the unit 20 is then set to cause that unit to act as a compressor, rather than as a motor. The handles 366 and 366′ are actuated while the controller 15′ is temporarily moved to its fully brake released position. The controller 15′ is then brought back to a position corresponding to the position of the then speed of the load. As the load accelerates, or tends to accelerate, it forces more and more air into the high pressure main, which air is taken from the main by the unit 363 operating as a motor. If the train should accelerate notwithstanding this braking action, one of two things will happen. Either the pressure will build up to the maximum value as determined by the safety valve 13 and then bleed from the high pressure main to the low pressure main through the safety valve or, as the pressure builds up and the braking effect increases the train will decelerate to a new speed as determined by the permitted rate of air out flow through the unit 363. If the pressure builds up to more than 600 pounds per square inch absolute, which is the assumed setting of the safety valve, it is desirable to set the controller to release more of the unloaders of the unit 363 to take more air from the high pressure main thus increasing the regenerative action and avoiding wastage of energy by the transfer of air through the safety valve. More cylinders of the unit 363 can be brought into action by shifting the hand lever of the controller 15′ to actuate more of the pilot valves 282 to release more of the unloader valves 275. If the rate of air consumption by the unit 363 as determined by the number of its released unloader valves is greater than the rate of air delivery by the unit 20 to the high pressure main, the pressure in that main will tend to drop, permitting an acceleration of the train to increase the rate of air delivery to the high pressure main until an equilibrium point is reached which will then determine the amount of regenerative braking present. On the other hand, if the rate of air consumption by the unit 363 is less than the rate of air delivery to the high pressure main by the unit 20, the pressure will tend to rise which will automatically increase the regenerative braking effort and decelerate the train to a new speed, at which the rate of air delivery by the unit 20 to the high pressure main exactly equals the rate of air consumption by the high pressure main.

Assume now that a load, such as a train, is operating at a level track and it is desired to use the regenerative braking to decelerate or stop the train. The handle of the controller 15′ is temporarily moved to the end position where the brake release piston 291 is fully depressed. The reversing cylinders 268—268′ are reversed, and then the controller is brought back to a position the same as the position it previously occupied. In that position of the controller the slightest deceleration of the train will cause its engine 20 to deliver less air to the unit 363 and thus the braking effect would be less. The controller is then moved back one step to reduce the number of cylinders of the engine 363 in service. This causes the unit 363 to take less air than is delivered to the high pressure main by the engine 20, with a result that the pressure in the main 360 tends to rise and therefore this pressure drives the unit 363 as a motor to drive the motor 362 as a generator for regenerative braking. As this regenerative braking continues the train decelerates until ultimately it reaches a speed equal to the speed set for it by the controller 15′. At this time all of the air supplied by the unit 20 flows to the unit 363 to operate it for regenerative braking. As the train decelerates further and decreases the rate of air supplied to the main 360, without a corresponding decrease in the rate of air consumption by the unit 363, the pressure in the main 360 commences to drop thus reducing the braking effort. The controller 15′ is then moved back one step to decrease the number of cylinders of the unit 363 taking air from the main 360, which again causes the pressure in the main 360 to rise. This action is continued, the controller 15′ being progressively moved back towards the neutral position as the train continues to decelerate.

From the above description it is apparent that the system of Figure 40 may be used for regenerative braking of other types of loads such as, for instance, a hoist.

In the power systems thus far described the maximum and minimum design pressure limits are so chosen that the maximum temperatures are limited. With an initial outside air temperature of 60° F. and a pressure range from 100 pounds per square inch absolute to 600 pounds per square inch absolute the temperature range, under adiabatic conditions, will be from 60° F. to 410° F. If desired the system may be designed to operate at a pressure cycle from 300 pounds per square inch absolute in the low pressure main to 600 pounds per square inch absolute in the high pressure main. With such a pressure range the temperature range during adiabatic compression and during adiabatic expansion, assuming an initial air temperature of 100° F., will be between 100° F. and 220° F. This is an appreciably lower temperature range than obtained when operating through a pressure cycle between 100 pounds per square inch absolute and 600 pounds per square inch absolute. Therefore, when operating between 300 pounds per square inch absolute and 600 pounds per square inch absolute it is possible to add heat from an external source to raise the temperature of the air in the high pressure main another 190° before a temperature of 410° F. is reached. An increase in temperature from 220° F. to 410° F. is the same as an increase from 680° F. absolute to 870° F. absolute. Assuming that, while heat is added from the external source, the pressure is unchanged by permitting an increase in volume, the volume will be increased by this heat addition in the ratio of 68 to 87. Hence, if this system of heating is used in addition to the compression from 300

pounds per square inch absolute to 600 pounds per square inch absolute, the capacity of the air compressor unit can be reduced in the same ratio, or can be reduced to 78% of the capacity that is required where heat from an external source is not added. Against the saving in capacity of the air compressor unit there must be debited the cost of equipment for raising the air temperature after compression and equipment for cooling the air temperature to 100° F. after it has been expanded in the compressed air engine.

While the above description has been directed to a system operated between a variable high pressure and a fixed low pressure, it is possible to operate the system between a fixed high pressure of 600 pounds per square inch and a fixed low pressure of, say, 100 pounds per square inch, or 300 pounds per square inch, as pointed out above. In a system designed to operate between fixed upper and lower pressures the power output of the engine is changed by changing the point of admission cut-off. The exhaust valve of the engine would then consist of an ordinary flutter valve which would open as soon as the air in the engine cylinder has been expanded to the fixed minimum value, namely, the pressure in the low pressure main.

As an alternative to either of the above arrangements it is possible to operate the system of Figure 1 between a fixed high pressure of, say, 600 pounds per square inch and a variable low pressure. Under those conditions the high pressure main would be of a comparatively high volumetric capacity and the low pressure main would be of a comparatively small volumetric capacity. The admission valves of the engine of Figure 2 would be operated to give a cut-off at 24%, which is the cut-off at 600 pounds as determined from Figure 7. By changing the point of opening of the exhaust valves it is possible to get the desired low pressure range. Under such circumstances the constant pressure maintaining apparatus of the auxiliary compressor 10 of Figure 1 would be omitted. Since the volumetric capacity of the low pressure main would be made very small the pressure in the low pressure main would be determined by the point of opening of the exhaust valves. If, for instance, the exhaust valve is opened after the gas in the cylinder has been expanded from 600 pounds to only 500 pounds, the remaining energy of the gas would not be lost because the pressure in the low pressure main would quickly rise to 500 pounds and thus reduce the amount of work required of the compressor unit to raise the pressure to 600 pounds.

In the description of the system illustrated in Figure 40 it was shown that air under pressure can be transmitted from the unit 363 to the unit 20 in variable amounts and that by proper setting of the reversing devises air can be delivered by the unit 20 to the unit 363 in variable amounts and that each one of the units 363 and 20 can operate either as a compressor when it is mechanically driven, or as a motor when it receives air under pressure. The load 21 can be replaced by a source of power to drive the shaft 33, operating the unit 20 as a compressor to deliver air to the unit 363 operating as a motor to drive a load. If the unit 20 is driven at a constant speed the amount of air delivered thereby into the high pressure main is constant. The speed of the unit 363, as previously described, will be determined by the number of its cylinders in service as determined by the position of the handle on the controller 15'. It has previously been pointed out how the handle of the controller 15' may be manipulated to give a constant speed for the unit 363 at a variable air intake. The controller 15' may also be manipulated to give a variable speed of operation of the unit 363 when the amount of air supplied thereto by the unit 20 is constant. Assume that the unit 20 is driven at a constant speed, delivering a constant amount of air to the high pressure main, and the unit 363 is driving a load which may be operated at different speeds such as, for instance, a hoist, or a locomotive load. Under such circumstances when the controller 15' is set to release the unloader valve of only one cylinder of the unit 363 the amount of air taken by the unit per revolution of its crank shaft is small, hence the pressure in the main 364 will build up thereby increasing the speed of the unit 363 until its speed is such that the rate at which it withdraws air from the main 364 equals the rate at which air is delivered to that main by the unit 20. If the controller is then set to release the unloader valves of two cylinders the unit 363 will thereupon commence to take twice as much air per revolution thereof and the pressure in the main 364 will therefore commence to drop, thereby dropping the speed of the unit 363 until new equilibrium conditions are reached, at half the speed that prevailed when only one cylinder is in service. Thus, progressively increasing the number of cylinders of the unit 363 that are in service will progressively decrease the speed of the unit 363, in a manner which is apparent from the description previously given. From the above description it may be seen that in a system employing two units such as the unit 363 and the unit 20, either unit may be driven at a constant speed to drive a load at a variable speed, and that the variations in the speed of the load may be obtained by releasing unloader valves on the unit that is acting as a motor or on the unit that is acting as a compressor.

The relationships of the speeds of the units 20 and 363 may be expressed mathematically as follows: Assume that S_1 and S_2 are the speeds of the units 363 and 20, respectively, and that N_1 and N_2 are the number of cylinders in service in the units 363 and 20, respectively. The product of S_1 and N_1 bears a direct relationship to the amount of air flowing through the machine 363 and the product of S_2 and N_2 bears a direct relationship to the amount of air flowing through the machine 20. Since the amount of air flowing through the two machines is the same, it follows that the product of S_1 and N_1 is directly proportionate to the product of S_2 and N_2. The following equation may therefore be written:

$$S_1N_1 = kS_2N_2$$

in which k is a constant. Therefore

$$S_1 = k\frac{S_2 N_2}{N_1}$$

or

$$S_2 = k\frac{S_1 N_1}{N_2}$$

It is therefore apparent that the speed of either machine can be controlled either by controlling the speed of the other machine or by controlling the number of cylinders in operation on either or both of the machines. The number of cyl-

inders in operation in either one of the machines may be held constant and the ratio

$$\frac{N_1}{N_2}$$

changed by actuating or releasing more or less of the unloaders of the other machine.

It is apparent that the principles of the present invention are of wide applicability and are particularly suited to the power requirements for the draw works of an oil well drilling rig or for the requirements for driving a vehicle such as, for instance, a train of cars, an automobile, or seacraft. In the case of a unit used for driving an automobile the present system dispenses with the usual clutch, speed changing gears, reversing gears, and foot brake.

As previously pointed out the air circuit of the present system is a closed circuit, the same air circulating over and over again through the high pressure main, the motor, the low pressure main, and the compressor, back to the high pressure main. The medium "air" is used because that is the most readily available gas. However, a different gas may be used if desired, in which case the auxiliary compressor 10 would take gas not from the atmosphere but from a suitably provided source. As previously pointed out, when operating at a low pressure of 100 pounds per square inch absolute the rise in temperature of the gas during compression limits the upper pressure to 600 pounds per square inch. With a gas other than air different upper limits may be utilized. It is therefore to be understood that other gases which can be compressed and expanded while remaining in their gaseous stage at the pressures and temperatures involved may be used as the equivalent of air in the system illustrated and claimed.

In compliance with the requirements of the patent statutes I have here shown and described a few preferred embodiments of my invention. It is, however, to be understood that the invention is not limited to the precise constructions here shown, the same being merely illustrative of the principles of the invention. What I consider new and desire to secure by Letters Patent is:

1. A power transmission including a high pressure air main, a low pressure air main, an air engine receiving air from the high pressure main and discharging into the low pressure main, an air compressor receiving air from the low pressure main and discharging into the high pressure main, means for maintaining the pressure in one of the mains constant and the pressure in both mains above atmospheric pressure whereby the engine operates between upper and lower pressures both above atmospheric pressure, said mains and compressor and engine forming a closed air system, and means for varying the pressure in the other main to vary the engine torque.

2. A power transmission including a high pressure air main, a low pressure air main, an air engine receiving air from the high pressure main and discharging into the low pressure main, an air compressor receiving air from the low pressure main and discharging into the high pressure main, means for maintaining the pressure in the low pressure main above atmospheric pressure whereby the engine operates between upper and lower pressures both above atmospheric pressure, said mains and compressor and engine forming a closed air system, means for varying the pressure in one of the mains to vary the engine torque, and means responsive to the pressure in said one main for regulating the engine valves in such relationship to the pressure in the high pressure main that the air in the engine cylinder during the compression stroke will have been compressed to the pressure of the high pressure main at the commencement of the opening of the intake valve.

3. A power transmission including a high pressure air main, a low pressure air main above atmospheric pressure, a reciprocating air engine having intake and exhaust valves receiving air from the high pressure main and discharging into the low pressure main, an air compressor receiving air from the low pressure main and discharging into the high pressure main, means for maintaining the pressure in one of the mains constant, means for altering the pressure in the other main, whereby the engine operates between a fixed and a variable pressure both above atmospheric pressure, means controlled by the pressure in the variable pressure main for regulating the termination of the engine exhaust in such relationship to the pressure in the variable pressure main that the air in the engine cylinder during the remainder of the exhaust stroke will have been compressed to the pressure of the high pressure main at the commencement of opening of the intake valve.

4. A power transmission system including a compressor unit, load driving means comprising an air engine of the type capable of operating also as a compressor, a high pressure main connecting the high pressure sides of the compressor and the engine, means for disabling the compressor unit when the engine is operating as a compressor to brake the load, throttling means to vary the pressure in the high pressure main, and manually operated brake regulating means for variably positioning the throttling means for variably discharging air from the high pressure main and thereby varying the pressure in the high pressure main and the braking effect of the engine on the load.

5. A power transmission system including a compressor unit, an air engine of the type capable of operating also as a compressor, a high pressure main connecting the high pressure sides of the compressor and the engine and a low pressure main connecting the low pressure sides of the compressor and the engine to constitute a closed air system, means for maintaining the pressure in the low pressure main substantially above atmospheric pressure, a controllable by-pass between the two mains, throttling means to vary the pressure in the high pressure main, and manually operated means for variably positioning the throttling means for variably opening the by-pass between the two mains to vary the pressure in the high pressure main and thereby vary the braking effect of the engine on the load.

6. A power transmission system including a compressor unit, an air engine of the type capable of operating also as a compressor, a high pressure main connecting the high pressure sides of the compressor and the engine and a low pressure main connecting the low pressure sides of the compressor and the engine to constitute a closed air system, and a controllable by-pass between the two mains, means for disabling the compressor unit when the engine is operating as a compressor, throttling means to vary the pressure in the high pressure main, and manually operated means for variably positioning the throttling means for variably opening the by-pass between

the two mains and thus to vary the pressure in the high pressure main thereby varying the braking effect of the engine on the load.

7. A power transmission including a high pressure air main, an air compressor unit discharging into the high pressure main, an air motor receiving air from the high pressure main at substantially the same rate as the rate of output of the compressor unit, at least the compressor unit including a plurality of cylinders, and manually operated means for changing the speed of the motor in relation to the speed of the compressor unit independently of the torque required of the motor, said means including means on a stationary portion of the said compressor unit for selectively disabling the cylinders thereof to alter the relationship of its speed to the speed of the motor and thereby increasing the maximum torque available to the motor.

8. In combination, a first air engine, a second air engine, a high pressure main connecting the high pressure sides of the two engines, each of said engines being of the type capable of operating by air pressure as a motor and operating as a compressor when mechanically driven, means coupled to the first engine to drive it as a compressor and to be driven by it when the said first engine operates as an air motor, said first engine having valve gear, means for reversing the valve gear to change said first engine from motoring to compressing while retaining the same direction of rotation, at least the first one of said engines comprising a multicylinder engine having manually operated means for changing the speed of the motor in relation to the speed of the compressor independently of the torque required of the motor, said means including means on a stationary portion thereof for selectively disabling the cylinders of the compressor unit to alter the relationship of its speed to the speed of the motor and thereby increasing the maximum torque available to the motor.

9. A power transmission including a closed circuit having a high pressure air main and a low pressure air main and means for maintaining the pressure in the entire circuit above atmospheric pressure, a multi-cylinder reciprocating air engine receiving air from the first main and discharging into the second main, said engine including mechanically operated intake and exhaust valves and having also means for maintaining the pressure within the engine cylinder between maximum and minimum limits substantially equal to the pressure in the respective mains.

10. A power transmission including a closed circuit having a high pressure air main and a low pressure air main and means for maintaining the pressure in the entire circuit above atmospheric pressure, a multi-cylinder reciprocating air engine receiving air from the first main and discharging into the second main, said engine including mechanically operated intake and exhaust valves and having also means for maintaining the pressure within the engine cylinder between maximum and minimum limits substantially equal to the pressure in the respective mains, said means comprising valves paralleling the mechanical valves and operated by the differences between the pressure in the cylinder and in the respective mains.

11. A power transmission including a closed system having high and low pressure air mains and means for maintaining the pressure in the entire circuit above atmospheric pressure and a multi-cylinder reciprocating engine receiving air from the high pressure main and discharging into the low pressure main, means responsive to a rise of pressure in the engine above that of the high pressure main for opening an air flow path between the engine cylinder and the high pressure main, and means responsive to a drop in pressure in the engine cylinder below that of the low pressure main for opening an air flow path between the engine cylinder and the low pressure main.

12. A power transmission including high and low pressure air mains and a reciprocating engine receiving air from the high pressure main and discharging into the low pressure main, means responsive to a rise of pressure in the engine above that of the high pressure main for opening an air flow path between the engine cylinder and the high pressure main, means mechanically operated by the engine piston for controlling the engine cut-off, means responsive to a drop in pressure in the engine cylinder below that of the low pressure main for opening an air flow path between the engine cylinder and the low pressure main, and additional means mechanically operated by the engine piston for controlling the termination of the engine exhaust action.

13. A power transmission including high and low pressure air mains and a reciprocating engine receiving air from the high pressure main and discharging into the low pressure main, means for varying the pressure in one of the mains to vary the engine output, means responsive to a rise of pressure in the engine above that of the high pressure main for opening an air flow path between the engine cylinder and the high pressure main, and means responsive to a drop in pressure in the engine cylinder below that of the low pressure main for opening an air flow path between the engine cylinder and the low pressure main.

14. A power transmission including high and low pressure air mains and a reciprocating engine receiving air from the high pressure main and discharging into the low pressure main, means for varying the pressure in one of the mains to vary the engine output, means responsive to a rise of pressure in the engine above that of the high pressure main for opening an air flow path between the engine cylinder and the high pressure main, mechanically operated means controlled by the engine piston for controlling the engine cut-off, means responsive to a drop in pressure in the engine cylinder below that of the low pressure main for opening an air flow path between the engine cylinder and the low pressure main, and additional mechanical means controlled by the engine piston for controlling the termination of the engine exhaust action.

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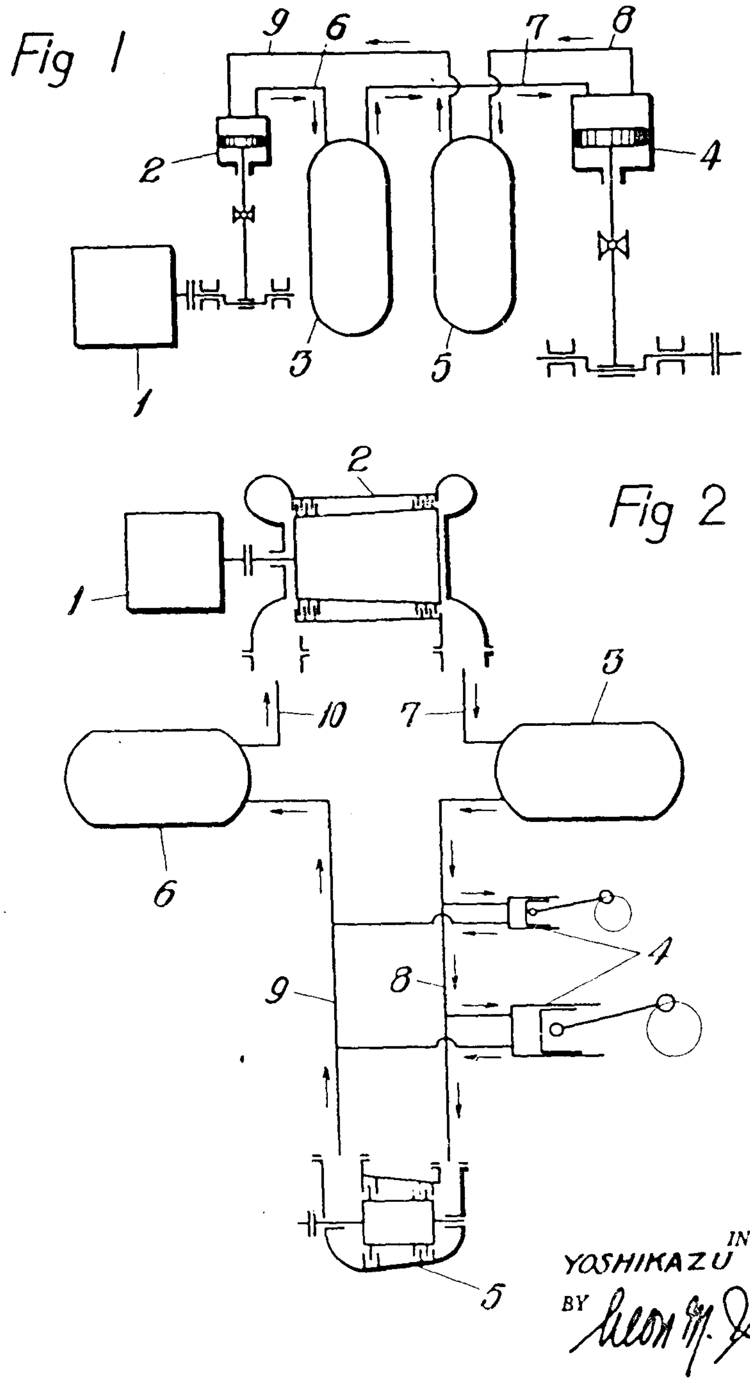
PNEUMATIC POWER TRANSMISSION SYSTEM

Filed March 22, 1957 3 Sheets-Sheet 1



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PNEUMATIC POWER TRANSMISSION SYSTEM

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Fig 3

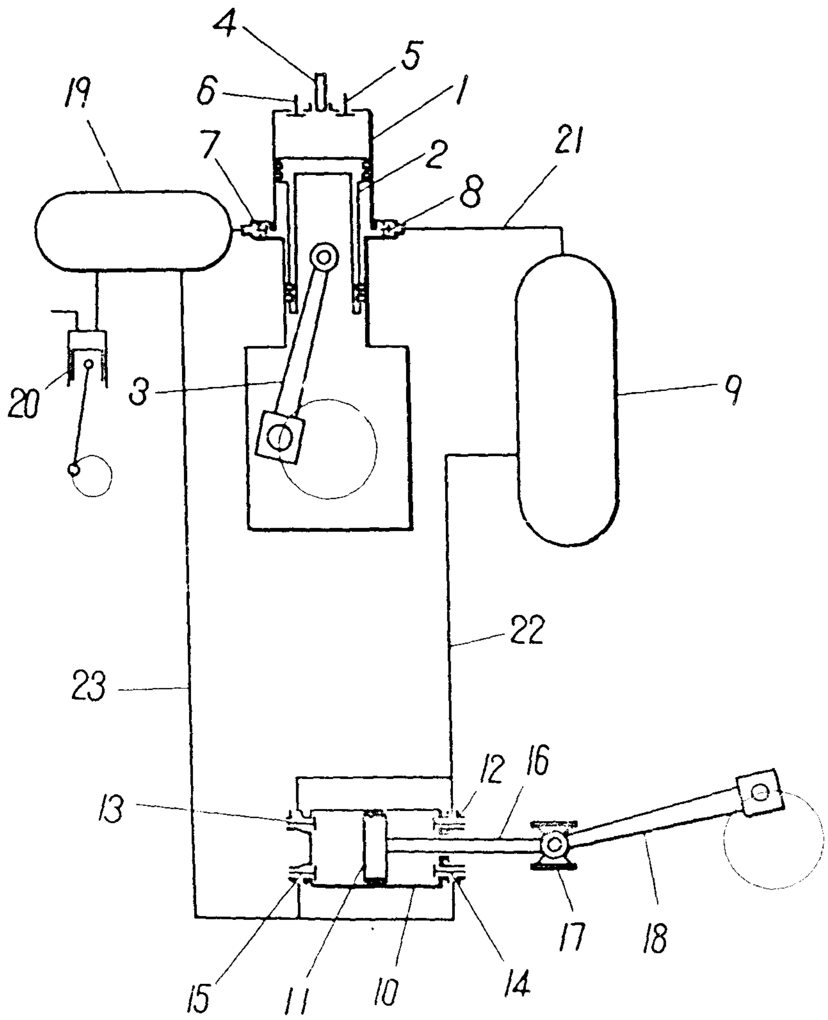

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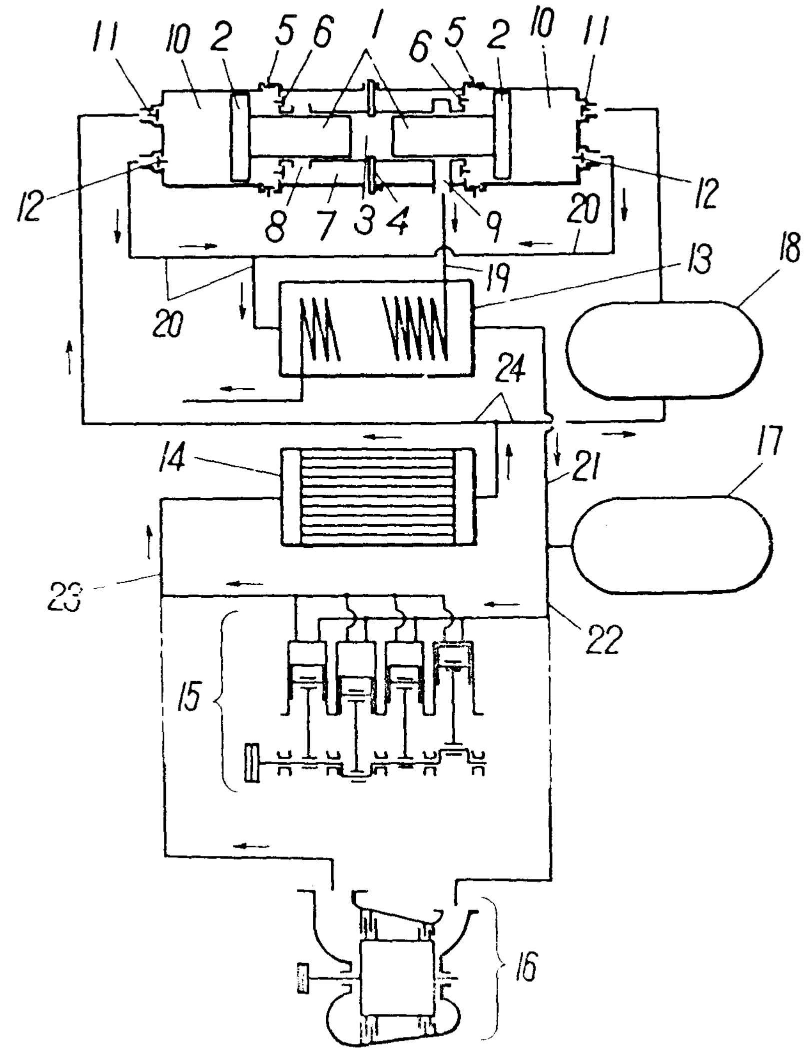
Fig 4



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PNEUMATIC POWER TRANSMISSION SYSTEM

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1 Claim. (Cl. 60—12)

The invention relates to a pneumatic transmission system using as its medium compressed air recirculating in a closed circuit.

Conventional pneumatic power transmission systems, in which one or more pneumatic motors are driven by compressed air to transmit power to the driven machine are normally of very low efficiency. This unfavourable efficiency is, first attributable to the fact that the adiabatic compression of air gives rise to a temperature increase therein. The heat quantity consumed in the temperature increase is, however, dissipated by radiation in the course of being conveyed through piping to the pneumatic motor, thus the compressed air acting as the working medium for the motor loses considerable energy and there is a reduction in the volume of the compressed air serving as the transmission medium, so that the energy to be spent in the pneumatic motor cylinder is correspondingly decreased. With increase of the compression ratio, this tendency will be further accentuated, as shown by the following data:

compression ratio	2	3	4	5	6	7
temperature rise, ° C	55	92	119	141	162	179
efficiency = output of pneumatic motor / power consumed by compressor	85	77	72	69	66	64

Next, the mechanical efficiencies of the compressor as well as of the pneumatic motor must be taken into account. Now assuming that the mechanical efficiency of the compressor amounts to 90% and that of the pneumatic motor be 80%, the resulting combined efficiency will be:

$$0.9 \times 0.8 = 0.72$$

that is to say, it amounts to 72%. When the above given numerical values listed are multiplied by 72%, the following values are obtained:

compression ratio	2	3	4	5	6	7
efficiency, percent	61	55	52	50	48	46

Third, the aqueous humidity in the air must be accounted for. When air is compressed, the humidity content contained therein will be liquidized and separated as so-called drain therefrom resulting merely in a loss. The higher compression ratio, the larger becomes the amount separated. Further, leakage, piping and other losses must be considered. When these losses are taken into account, the overall efficiency will be further decreased from those listed above.

In the pneumatic power transmission system according to the present invention, the exhaust port of the pneumatic motor is connected through a piping to the suction port of the compressor, thus providing a closed circuit for the power transmission medium. By this procedure, the suction pressure of the compressor amounts to several

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atmospheric pressures or a multiple thereof and thus a satisfactory pressure difference will be easily obtained with a low compression ratio. This allows the employment of a correspondingly smaller sized pneumatic motor and a lower compression ratio, thus avoiding the unfavorable efficiency caused by higher compression ratio. A smaller pneumatic motor provides a smaller area of sliding surface, which will, in turn, increase the efficiency by reducing friction loss.

Now assuming that a compression ratio of 2 or lower is employed, the temperature rise will amount to less than 55° C. With this temperature, the temperature difference relative to the atmospheric temperature is small, so that in this case the radiation loss is also insignificant. On the other hand, the compressor sucks in air having a relatively low temperature after dissipation of energy in the pneumatic motor cylinder. When these conditions are considered, the difference in temperature of the delivered air from the compressor relative to atmospheric temperature will amount to only about 20–30° C. This condition ensures a relatively lower radiation loss, which may be of the order of, say, 5%. If the necessary heat insulation is made in a satisfactory manner, the last mentioned energy loss will be further decreased.

Now turning to the mechanical efficiencies, it may be assured, that those of compressor and pneumatic motor are expected to be about 90% and 90%, respectively, based on the observation that the frictional surfaces are much smaller than in conventional systems of similar kind. The overall efficiency will be:

$$0.9 \times 0.9 \times (1 - 0.05) = 0.77$$

that is to say, 77%. On the contrary thereto, with the known system, the friction loss may amount to as high as 10%, because of the drain loss already explained hereinbefore: With a compression ratio 7, that is, with the compressed air, 7 kg./sq. cm., the overall efficiency taken into account of said friction loss will amount to:

$$0.46 \times 0.9 = 0.41 \text{ (or } 41\%\text{)}$$

With the present invention, which employs a closed circuit for the transmission medium, the same dry air is recirculated therethrough without such a drain loss as well as appreciable lubricating oil loss.

When an efficiency ratio between the novel and the known pneumatic power transmission system is taken, the following value is thus found:

$$77/41 = 1.88$$

which means a heavy increase in the transmission efficiency in the favor of the present invention.

In an electric power transmission using a generator-motor combination, when assumed:

	Percent
Efficiency of generator	90
Efficiency of motors:	
(when employed a plurality of smaller electric motors)	80
(or alternatively, when fewer larger electric motors are employed)	85

the required overall efficiency will be

$$0.9 \times 0.8 \sim 0.9 \times 0.85 \quad 0.72 \sim 0.77$$

This value means that the power transmission system according to this invention is almost equal in efficiency to the electrical system set forth above.

On the other hand, the system according to this invention can be more easily manufactured at a lower cost and provides a means for easier manipulation of speed within broader limits carried into effect by using a throttle valve, or alternatively by means of a suitable valve

mechanism. A further special feature of the present system resides in that there is no trouble caused by overheating, and the invention gives rise to compact design and the possibility of low speed running.

Various further and more specific objects, features and advantages of the invention will appear from the description given below, taken in connection with the accompanying drawings diagrammatically illustrating by way of example several embodiments of this invention.

In the drawings:

Fig. 1 shows a general arrangement of the first embodiment of the invention, in which the air compressor and the pneumatic motor are of the reciprocating type;

Fig. 2 represents a similar view to Fig. 1, showing the second embodiment of the invention, in which the compressor is an axial turbo-compressor and the pneumatic motor of the reciprocating type of the turbo-type, as the occasion may desire;

Fig. 3 shows the third embodiment of the invention, in which the engine and the compressor are united in a combined machine;

Fig. 4 represents the fourth embodiment of the invention, in which prime mover and compressor are constructed as a free piston type diesel engine-compressor unit and an air heater is provided, the latter being adapted to heat the compressed air delivered from the compressor by the exhaust gases discharged from the engine.

Now referring to the drawings there is shown a prime mover **1**, preferably a diesel engine, which is coupled with an air compressor of the reciprocating type, the compressed air delivered from the latter being conveyed through piping **6** to a high pressure air reservoir **3**. A pneumatic motor **4** of the reciprocating type is coupled with a suitable driven member, for instance, the driving axle of a locomotive. The working air for the motor **4** is delivered from the high pressure reservoir **3** through piping **7** and the discharged air from said motor is conveyed through piping **8** to a low pressure air reservoir **5** to be accumulated therein. The air is sucked by the compressor **2** through piping **9**. The pneumatic motor **4** is illustrated as larger in size than the compressor **2**, because of the fact that the motor **4** is adapted to drive the driven member at relatively lower revolutions and thus with a larger torque as the occasion demands.

For the sake of simplicity, the manipulating means inclusive the valve means arranged in the pneumatic circuit has been omitted from the representations in the present and following drawings. The mode of operation of the present transmission system would be clear to those skilled in the art, when they read the above explanation in combination with the introduction hereinbefore disclosed.

In the second embodiment of this invention shown in Fig. 2, **1** denotes again a prime mover, preferably a diesel engine, which is, however, in this case coupled with an axial flow turbo-compressor **2**, the delivery-side of which is connected through piping **7** to a high pressure air reservoir **3**. The working air for one or more pneumatic motors **4** of the reciprocating type is supplied from the latter through piping **8**. The crankshaft of each motor **4** is direct-connected to a member to be driven, for instance, a driving axle of a locomotive (not shown) as in the previous embodiment. The air discharged from said motors **4** is conveyed through piping **9** into a low pressure air reservoir **6** to be accumulated therein and thence returned to the low pressure side of said compressor **2**, thus completing the power transmission cycle. As an alternative measure, said one or more reciprocating type pneumatic motors **4** may be replaced by a pneumatic motor of the turbo-type, or compressed air turbine **5**, as shown in the lower part on Fig. 2. This alternative arrangement is especially suitable for driving of a machine having a higher running speed. These two arrangements are used to transmit a relatively large power to the driven machine. The air reservoirs arranged therein insure that the system will operate substantially at a predetermined pressure level in case of load fluctuations. Because of the fact that the pneumatic motor or motors employed in the present embodiment shown in Fig. 2 are of the reciprocating or alternatively of the rotary type, as the occasion may be, the power transmission will meet simply and satisfactorily the special speed characteristics of the driven machine at issue.

In the third embodiment of the invention shown in Fig. 3, **1** denotes one of the working cylinders of a diesel engine, in which a piston **2** reciprocates. The connecting rod **3** is operatively connected, as in the usual manner, with the crankshaft of the engine. The piston **2** is formed in a stepped piston, the underside of which serves as a compressor piston. Fuel injection valve **4**, suction valve **5** and exhaust valve **6** of the engine are arranged in the cylinder cover, while suction valve **7** and delivery valve **8** of the compressor are mounted on the combined cylinder **1**. The air delivered by the compressor is conveyed through piping **21** into a high pressure reservoir **9**, as in the case of precedent embodiments, to be accumulated therein. **10** represents a working cylinder of a pneumatic motor of the double-acting type, the suction sides of which are connected through piping **22** to said high pressure reservoir **9**. The piston **11** is arranged to reciprocate therein and connected through piston rod **16**, crosshead **17** and connecting rod **18** with a driven member, for instance, a driving axle of a locomotive. Suction valves **12**, **13** and exhaust valves **14**, **15** are arranged in the cylinder **10** in the manner known per se. The air discharged from said pneumatic motor is conveyed through piping **23** into a low pressure air reservoir **19**, to which is attached a small auxiliary air compressor **20** serving for replenishing possible leakage loss and thus maintaining the pressure prevailing in the low pressure circuit at a predetermined value.

In the present embodiment, the engine cylinder is combined with that of the air compressor as already mentioned, the piston being formed in a stepped one. Further, the compression ratio of the compressor is selected at a relatively lower value, the pressure in the low pressure circuit amounting to 10 kg./sq. cm. or higher. The last mentioned feature makes it possible to employ a higher mean effective pressure of the compressor and thus a small size piston or pistons therefor. Based upon this feature, the utilization of underside of the engine piston as the compressor piston is realized in the present embodiment.

The working mode of the above mentioned system is as follows:

Air is charged by the auxiliary air compressor **20** into the low pressure reservoir **19** till a pressure of, say 10 kg./sq. cm. is reached. Then, the diesel engine is brought into running, thus the combined air compressor further compressing the air coming from said reservoir **19** to an elevated pressure, say 20 kg./sq. cm. and charging it into the second reservoir **9**. The air is thence delivered through piping **22** and suction valves **12**, **13** to motor cylinder **10**, by which the double acting piston **11** is reciprocated to and fro in said cylinder. Power is thus transmitted through piston rod **16**, crosshead **17** and connecting rod **18** to the driven member, for instance, a driving wheel of a locomotive. The expanded air in the cylinder **10** is discharged therefrom through delivery valves **14**, **15** and piping **23** into the first or low pressure reservoir **19**, thus completing the transmission cycle.

In the present embodiment, no power is derived from the crankshaft of the engine-compressor combination and the crankshaft serves mainly for transmitting the compressive forces caused by a set of pistons arranged in a row, so that the shaft may be of smaller size and simpler construction as compared with that in a corresponding diesel engine of the same output. The crank-

shaft in this case may be more easily manufactured at a lower cost and have practically no troubles caused by torsional vibrations, shaft breakage and the like as in the normal plant.

In the fourth embodiment of the invention illustrated in Fig. 4, 1 denotes the opposed pistons of a free piston type diesel engine and 2 represents air compressor pistons united therewith, thus constituting two stepped pistons. The outer sides of the larger pistons 2 serve as main compressor pistons, while the inner sides of the pistons 2 act as the scavenging air compressor ones. The engine pistons 1 reciprocate in the main cylinder 3, which is provided with a plurality of fuel injection nozzles 4. Each of the scavenging compressor cylinders is provided with a suction valve 5 as well as with a delivery valve 6 as in the usual manner. A scavenging air cylinder 7 is arranged around the main engine cylinder 3 and made integral with said scavenging compressor cylinder. A plurality of scavenging ports 8 as well as exhaust ports 9 are arranged, the latter leading through piping 19 to an air heater 13. The compressor cylinders 10 serve, as above mentioned, for the main compressors for working air as well as for the auxiliary compressors to deliver the scavenging air for the engine. The main compressors are, as known per se, provided with suction and delivery valves 11 and 12 in the cylinder covers. The delivery sides of the main compressors are connected through piping 20, air heater 13 and piping 21 to a high pressure air reservoir 17, which is, in turn, connected through piping 22 to the suction side of a pneumatic motor of the reciprocating type 15 or alternatively of the turbo-type 16, depending upon the speed characteristics of the machine to be driven (not shown). The air discharged from the motor 15 or 16 is conveyed through piping 23 to an intercooler 14, which is connected through piping 24 by way of a lower pressure air reservoir 18 or directly to the suction sides of the main compressors of the main cylinder.

The mode of operation of the above mentioned embodiment is as follows:

When the engine pistons 1 move inwards to compress the air supplied through scavenging ports 8 and arrive nearly at their inside dead points, the fuel is supplied through the fuel injection nozzles 4 into thus highly heated air in the engine cylinder 3, resulting in a combustion of the fuel. Thus generated combustion gases expand and drive the engine pistons outwards to initiate the expansion stroke. When the pistons are brought nearly at their outside dead points, the exhaust ports 9 open to discharge the combustion gases and then the scavenging ports 8 are opened to introduce the scavenging air accumulated in the air chamber 7 and thereby to drive the residual gases out from the cylinder through exhaust ports 9, thus the combustion gases being almost completely replaced by the fresh air of a relatively lower pressure. During the expansion stroke of the engine, energy is transmitted from the combustion gases through engine pistons 1 to compressor pistons 2, which then compress the air in the main compressor chambers 10 and deliver it through delivery valves 12. When the pistons arrive at their outside dead points, the compressed air contained in the clearance spaces of the compressor cylinders 10 pushes them back towards their inside dead points to initiate again the compression stroke of the engine, the movement of the pistons being further assisted by the influence of the introduced compressed air through inlet valves 11 from the low pressure circuit of the system in the course of this compression stroke. Although not shown in the drawing, there is provided a suitable synchronizing mechanism, such as linkage or rack and pinion mechanism to insure the correlated, properly timed relations between both side pistons. The air compressed by the main compressor pistons 2 in the cylinders 10 is conveyed through delivery valves 12 and piping 20 to air heater 13, wherein it is heated by the

exhaust gases discharged from the engine cylinder 3. On the other hand, the air introduced through suction valves 5 from the atmosphere into the scavenging air cylinders during the expansion stroke of the engine, is compressed by the opposite sides of said pistons 2 during the compression stroke of the engine and delivered through delivery valves 6 into the air chamber 7 to be accumulated therein, said air serving as the scavenging air for the engine already explained hereinbefore. The compressed air heated, as abovementioned, in the air heater 13 increases in its temperature and pressure, and, after accumulated in the high pressure reservoir 7 if necessary, carries out the necessary work, when it expands in the pneumatic motor 15 or 16 to drive the machine to be driven, thus completing the power transmission. The discharged air from said motor has a relatively higher temperature, resulting from the main feature of the present system, which operates with a relatively lower compression ratio of the main compressors with a lower temperature drop during the adiabatic expansion. The relatively hot discharged air is then cooled in the intercooler 14 and thus cooled air to a substantial degree is thence supplied through inlet valves 11 into the main compressor cylinders 10 to initiate again the abovementioned cycle and so on.

Now assuming that the efficiency of the free piston type diesel engine be 35%, that of the main compressors 90% and that of the pneumatic motor of the reciprocating type 90%, the overall efficiency will substantially amount to:

$$0.35\times 0.9\times\frac{273+200}{273+100}=0.359$$

wherein the quotient represents an increase in efficiency obtained by the provision of said air heater, the temperatures of the air being assumed 100 and 200° C. at the inlet and the outlet, respectively. This value is somewhat higher than that found in the normal diesel engine. The overall efficiency will be somewhat decreased, when a turbo-type pneumatic motor is employed.

Although certain particular embodiments of the invention are herein disclosed for purpose of explanation, further modifications thereof, after study of this specification, will be apparent to those skilled in the art to which the invention pertains. Reference should accordingly be had to the appended claims in determining the scope of the invention.

What is claimed as new and desired to be secured by Letters Patent is:

A pneumatic power transmission system using compressed air recirculating in a closed circuit; comprising a free piston type diesel engine having a main cylinder, a compressor cylinder surrounding said main cylinder, a pair of opposed free pistons in said main cylinder, air compressor pistons fixed to said free pistons and being disposed in said compressor cylinder, means dividing the space between said main cylinder and said compressor cylinder to form compressor chambers at the ends of said compressor cylinder and scavenging air chambers about said main cylinder, scavenging air ports for connecting said scavenging air chambers with the interior of said cylinder, the air compressor pistons serving as main compressors delivering the compressed air as transmission medium and the free pistons being employed for producing scavenging air necessary for said engine, an air heater to heat the compressed air delivered from said main compressors by the exhaust gases discharged from said engine, a high pressure air reservoir, a pneumatic motor operatively connected with the member to be driven, said high pressure reservoir accumulating the air in the high pressure air circuit between said main compressors and said pneumatic motor and said motor being driven by the air in the last mentioned circuit, intercooler means arranged in the low pressure air circuit between said pneumatic motor and said main compressors,

and a low pressure air reservoir arranged in the last mentioned low pressure circuit when necessary, said low pressure air reservoir accumulating the air in the low pressure air circuit.

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US005957234A

United States Patent [19]

Manor

[11] **Patent Number: 5,957,234**

[45] **Date of Patent: Sep. 28, 1999**

[54] **COMPRESSED AIR POWERED MOTOR VEHICLE**

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[21] Appl. No.: **09/048,515**

[22] Filed: **Mar. 26, 1998**

[51] **Int. Cl.**^{6} **B60K 9/00**

[52] **U.S. Cl.** **180/302**; 60/412

[58] **Field of Search** .... 180/305, 306, 180/307, 308, 302, 301; 60/712, 715, 719, 407, 412; 123/198 C

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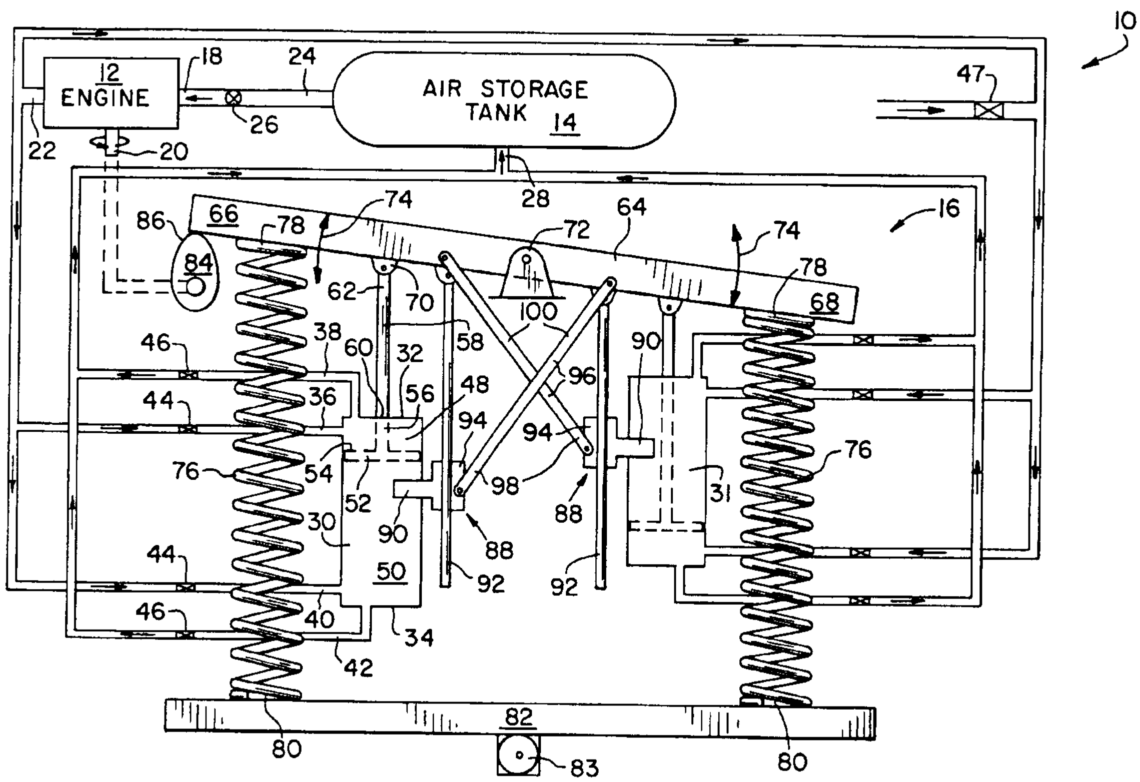
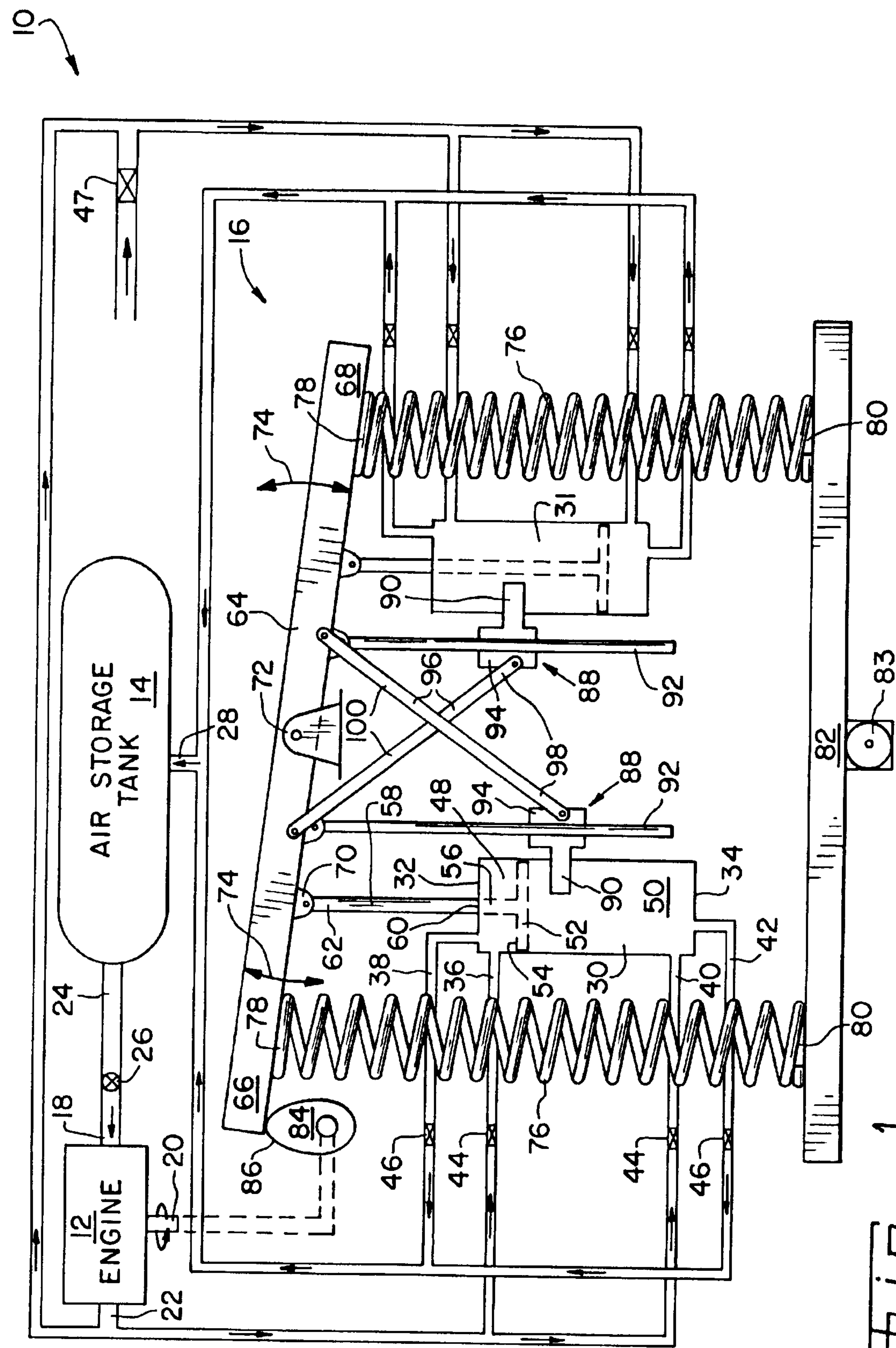
Primary Examiner—Richard M. Camby
Attorney, Agent, or Firm—Taylor & Associates, P.C.

[57] **ABSTRACT**

A compressed air powered motor vehicle includes an engine having an intake and an exhaust pipe. The engine is configured for operating on compressed air received in the intake and expelling exhaust air through the exhaust pipe. A compressed air storage container is in fluid communication with the intake of the engine. At least one compression mechanism is in fluid communication with the exhaust pipe of the engine and with the compressed air storage container. The at least one compression mechanism is configured for compressing the exhaust air and replenishing the compressed air storage container. Each compression mechanism includes a reciprocable connecting rod having a direction of travel. A rocker arm has a first end, a second end and a pivot point disposed between the first end and the second end. The first end of the rocker arm is attached to the connecting rod. The rocker arm is configured for pivotal oscillation about the pivot point, thereby reciprocating the rod within the compression mechanism. A bracket interconnects the second end of the rocker arm and the cylinder. The bracket carries the cylinder and moves the cylinder in a direction opposite to the direction of travel of the rod. A stabilizing bar is oriented substantially parallel to the rod and is configured for slidably carrying the bracket.

14 Claims, 2 Drawing Sheets




AIR STORAGE
TANK 14
12
ENGINE
10
16
18
20
22
24
26
28
30
31
32
34
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38
40
42
44
46
47
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78
80
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86
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92
94
96
98
100

Fig. 1

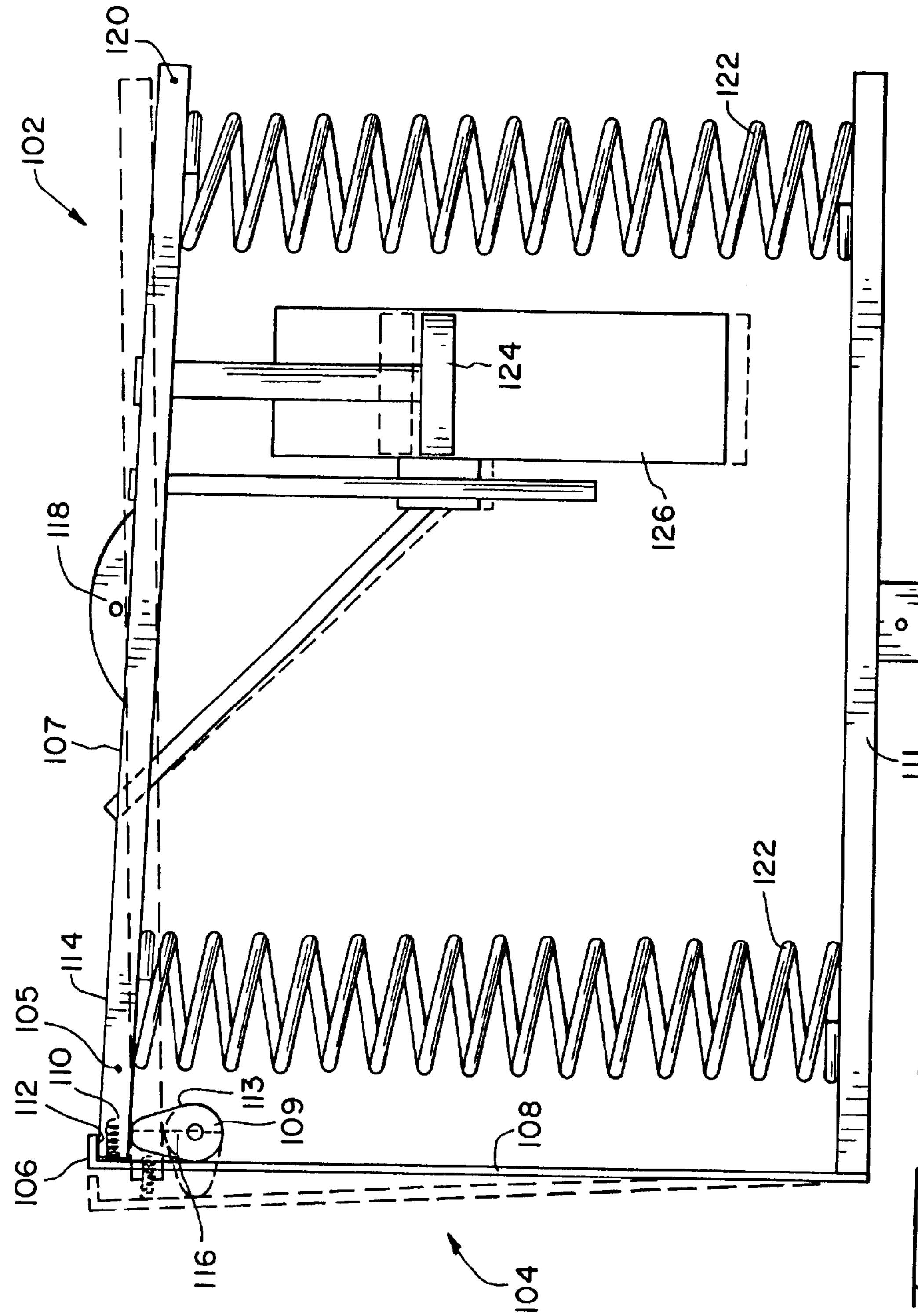
120
102
122
124
126
118
107
111
122
114
105
110
112
113
109
108
106
116
104

Fig. 2

COMPRESSED AIR POWERED MOTOR VEHICLE

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to vehicles, and, more particularly, to a vehicle that operates from gaseous fluid such as air under pressure.

2. Description of the Related Art

An air powered vehicle, having a chassis and wheels, includes an air powered engine mounted on the chassis and having a driving connection with the wheels. A reservoir of gaseous fluid under pressure is connected to an intake system for operating the engine. The air powered engine also includes an exhaust system for expelling still partially compressed exhaust air. Such air powered vehicles are disclosed in U.S. Pat. No. 3,847,058 (Manor) and U.S. Pat. No. 3,980,152 (Manor).

It is known to recompress exhaust air from an air powered engine using a battery operated compressor and return the recompressed air to an air storage tank. A problem is that a conventional 12 volt battery is capable of storing only a very limited amount of power. Although it is possible to recharge the battery using energy from the engine, such recharging involves substantial energy losses and is generally inefficient. Using a great number of batteries to power the compressor is also not practical, as the batteries are expensive and heavy, thereby reducing the overall efficiency of the vehicle.

It is also known to use the relative vertical motion between the chassis and the axle or wheels to recompress the exhaust air using a second type of compressor which is designed to be driven by the vertical motions of the vehicle. A problem is that the additional expense and weight of this second type of air compression system may not be justified, as the energy recoverable from the vertical motions of the vehicle may be very limited, especially on smooth roads.

What is needed in the art is an air powered vehicle in which exhaust air is recompressed using a compression mechanism driven directly by the air powered engine.

SUMMARY OF THE INVENTION

The present invention provides a compressed air powered motor vehicle including an engine which directly drives a compression mechanism for recompressing exhaust air and returning it to a compressed air storage container.

The invention comprises, in one form thereof, a compressed air powered motor vehicle including an engine having an intake and an exhaust pipe. The engine is configured for operating on compressed air received in the intake and expelling exhaust air through the exhaust pipe. A compressed air storage container is in fluid communication with the intake of the engine. At least one compression mechanism is in fluid communication with the exhaust pipe of the engine and with the compressed air storage container. The at least one compression mechanism is configured for compressing the exhaust air and replenishing the compressed air storage container. Each compression mechanism includes a reciprocable connecting rod having a direction of travel. A rocker arm has a first end, a second end and a pivot point disposed between the first end and the second end. The first end of the rocker arm is attached to the connecting rod. The rocker arm is configured for pivotal oscillation about the pivot point, thereby reciprocating the rod within the compression mechanism. A bracket interconnects the second end of the rocker arm and the cylinder. The bracket carries the cylinder and moves the cylinder in a direction opposite to the direction of travel of the rod. A stabilizing bar is oriented substantially parallel to the rod and is configured for slidably carrying the bracket.

An advantage of the present invention is that the output of the engine can be used to directly drive an air compression mechanism, thereby maximizing the overall efficiency of the vehicle.

Another advantage is that batteries are not needed to recompress the exhaust air of the engine.

Yet another advantage is that exhaust air may be recompressed under any road conditions.

BRIEF DESCRIPTION OF THE DRAWINGS

The above-mentioned and other features and advantages of this invention, and the manner of attaining them, will become more apparent and the invention will be better understood by reference to the following description of embodiments of the invention taken in conjunction with the accompanying drawings, wherein:

FIG. **1** is a schematic diagram of one embodiment of a compressed air powered motor vehicle of the present invention; and

FIG. **2** is a schematic diagram of another embodiment of the compression mechanism of the compressed air powered motor vehicle of FIG. **1**.

Corresponding reference characters indicate corresponding parts throughout the several views. The exemplifications set out herein illustrate one preferred embodiment of the invention, in one form, and such exemplifications are not to be construed as limiting the scope of the invention in any manner.

DETAILED DESCRIPTION OF THE INVENTION

Referring now to the drawings and particularly to FIG. **1**, there is shown a compressed air powered motor vehicle **10** including a compressed air powered engine **12**, a compressed air storage container **14** and an air compression mechanism **16**.

Engine **12** is powered by compressed air received from storage container **14** through an intake **18**. Engine **12** converts the energy of the compressed air into a rotation of an output shaft **20**. The operation of such an air powered engine is well known and is not discussed in detail herein. Exhaust air, which may be partially compressed as compared to ambient air, is expelled through an exhaust pipe **22**.

Air storage tank **14** feeds compressed air into intake **18** of engine **12** through an outlet **24**, regulated by a valve **26**. Storage tank **14** receives recompressed air through an inlet **28**. Storage tank **14** is of a strength so as to contain air at approximately 30 to 500 p.s.i.

Air compression mechanism **16** includes, as shown in FIG. **1**, a left-hand cylinder **30** and a right-hand cylinder **31**. The two cylinders are substantially identical, and hence only the left-hand cylinder **30** is referred to in detail herein. Cylinder **30** is substantially hollow and has two opposite ends **32** and **34**. End **32** has an inlet **36** and an outlet **38**, and end **34** has an inlet **40** and an outlet **42**. Each of inlets **36** and **40** is in fluid communication with exhaust pipe **22** through a respective one-way check valve **44**. Each check valve **44** allows passage of the exhaust air into cylinder **30** while preventing passage of the exhaust air out of cylinder **30**. Similarly, each of outlets **38** and **42** is in fluid communica-

tion with inlet **28** through a respective one-way check valve **46**. Each check valve allows passage of compressed air out of cylinder **30** while preventing passage of the compressed air into cylinder **30**. A one-way check valve **47** allows passage of ambient air into check valves **44** in case an adequate supply of air is not available from exhaust pipe **22**.

Cylinder **30** includes a first chamber **48** and a second chamber **50** separated by a piston **52**. Piston **52** forms a substantially airtight seal between first chamber **48** and second chamber **50**. First chamber **48** is in fluid communication with inlet **36** and outlet **38**, while second chamber **50** is in fluid communication with inlet **40** and outlet **42**. Piston **52** is slidable along a portion of the length of cylinder **30**, and maintains, even while sliding, the substantially airtight seal between chambers **48** and **50**. A first side **54** of piston **52** is attached to a first end **56** of a connecting rod **58** which extends axially from cylinder **30**. Rod **58** is slidable through an orifice **60** in first end **32** of cylinder **30**, forming a substantially airtight seal therewith. A second end **62** of rod **58** remains disposed outside of cylinder **30**. A rocker arm **64** has two opposite ends **66** and **68**, each of which is pivotally connected to a second end **62** of a respective rod **58** through a respective pivot **70**. Ends **66** and **68** of rocker arm **64** are separated by a pivot point **72** about which rocker arm **64** may pivot, as indicated by arrows **74**. Pivot point **72** can be in the form of a pillow block bearing. A respective suspension spring **76** supports each of ends **66** and **68** of rocker arm **64**. One end **78** of each suspension spring **76** is attached to rocker arm **64**, while a second end **80** of suspension spring **76** is attached to a fixed structure **82**. Fixed structure **82** is shown as being supported by an axle **83**.

A non-circular, substantially oval cam **84** is coupled to end **66** of rocker arm **64**. End **66** is biased against an outside surface **86** of cam **84** by the attached spring **76** as well as the weight of rocker arm **64**. Cam **84** is carried and driven by crank shaft **20** of engine **12**.

Each of two brackets **88** interconnects a respective cylinder **30** with a respective opposite end of rocker arm **64**. For instance, the left-hand bracket **88** of FIG. **1** interconnects a cylinder **30** to end **68** of rocker arm **64**, end **68** being opposite from end **66** of rocker arm **64**, to which a cylinder **30** is connected through a rod **58**. Brackets **88** each include two arms **90** (only one of which can be seen in FIG. **1**) which are rigidly attached to cylinder **30** and enable bracket **88** to movably carry cylinder **30**.

Two elongate stabilizer bars **92** are each oriented substantially parallel to a corresponding rod **58**, and each stabilizer bar **92** slidably carries a respective bracket **88**. Each stabilizer bar **92** is pivotably attached to rocker arm **64**. Each bracket **88** includes a body **94** which substantially encloses an associated stabilizer bar **92**. It is also possible to provide two helical springs (not shown) surrounding and concentric with each stabilizer bar **92**, an end of each spring contacting an opposite end of body **94**. The springs can be held fixed on their respective opposite outside ends, thereby spring-loading each body **94** for smoother sliding movement. Each of two connecting bars **96** has opposite ends **98** and **100** which are pivotally connected to a bracket body **94** and an end of rocker arm **64** opposite thereto, respectively.

During use, engine **12** rotates cam **84** through output shaft **20**. As cam **84** rotates, end **66** of rocker arm **64** rides on the substantially oval outside surface **86** of cam **84**. Rocker arm **64** pivotally oscillates about pivot point **72** as end **66** follows surface **86** of rotating cam **84**. The pivotal oscillation of rocker arm **64** causes rod **58** and piston **52** to reciprocate back and forth, piston **52** oscillating up and down within cylinder **30**. As end **68** pivots downwardly, opposite end **66** pivots upwardly, pulling the left-hand rod **58** and piston **52** upwardly with it. The connecting bar **96** that is attached to the left-side bracket **88**, on the other hand, follows the downward movement of end **68** and carries its bracket **88** and the corresponding cylinder **30** downward, opposite to the direction of travel of its piston **52**. Conversely, as end **66** pivots downwardly and pushes left-hand rod **58** and piston **52** down with it, oppositely pivoting end **68** pulls left-side bracket **88** and cylinder **30** upwardly, again opposite to the direction of the corresponding piston **52**. Stabilizer bars **92** guide and confine the movement of brackets **88** and cylinders **30**, keeping the movement of cylinder **30** substantially parallel to its rod **58**.

As piston **52** moves upwardly with respect to its associated cylinder **30**, air within first chamber **48** is compressed. When the air pressure within first chamber **48** is greater than the air pressure within storage tank **14**, check valve **46** within outlet **38** opens and allows the compressed air to be transferred to storage tank **14**. Conversely, as piston **52** moves upwardly, a negative air pressure or vacuum is drawn on second chamber **50**, which results in exhaust air being drawn from exhaust pipe **22** through check valve **44** of inlet **40**. When piston **52** again moves downwardly with respect to cylinder **30**, the air which was previously drawn into second chamber **50** is compressed by piston **52** and pushed out of outlet **42** through check valve **46** to replenish air storage tank **14**. During this downward movement of piston **52**, exhaust air is drawn into first chamber **48**, substantially identically to the way exhaust air was drawn into second chamber **50** as described above. When the air pressure at exhaust pipe **22** exceeds the air pressure within first chamber **48**, check valve **44** of inlet **36** opens and allows exhaust air to flow into chamber **48**. In this way, compressed air is expelled through outlet **38** on an upstroke, and is expelled through outlet **42** on a downstroke.

In an alternative embodiment (FIG. **2**), an air compression mechanism **102** includes a spring lock mechanism **104** which holds end **105** of rocker arm **107** stationary in the event that cam **109** becomes stationary. Spring lock mechanism **104** includes a latch **106** connected to fixed structure **111** by an elongate element **108**. A locking spring **110** interconnects latch **106** and end **105** of rocker arm **107** and biases latch **106** against outside surface **113** of cam **109**. Latch **106** includes a bottom side **112** which engages a top side **114** of rocker arm end **105** when a longitudinal extension **116** of cam **109** is substantially perpendicular to rocker arm **107**. Rocker arm end **105** is thereby clamped between bottom side **112** of latch **106** and cam **109**. With no other forces being applied to cam **109**, the bias of locking spring **110** is sufficient to pull elongate element **108** against cam **109**, thereby holding cam **109** substantially stationary such that its longitudinal extension **116** remains substantially perpendicular to rocker arm **107**. In this embodiment, a pivot point **118** can be detachable so that rocker arm end **105** can function as a pivot point about which rocker arm **107** may oscillate. Alternatively, pivot point **118** may be attached to another spring (not shown) such that pivot point **118** may have some vertical movement, allowing end **105** to function as a pivot point for rocker arm **107**.

During use, it is possible for cam **109** to no longer be driven by engine **12**, e.g., while vehicle **10** is coasting and engine **12** is turned off. In this scenario, the bias of locking spring **110** will pull latch **106** against both end **105** and cam **109**, thereby locking longitudinal extension **116** of cam **109** substantially perpendicular to rocker arm **107** and clamping rocker arm end **105** between bottom side **112** of latch **106**

and outside surface **113** of cam **109**. Road anomalies and the resultant up and down movement of vehicle **10** will cause the free end **120** of rocker arm **107** to oscillate and pivot about the now pivotal end **105** under the stabilizing influence of springs **122**. The oscillation of rocker arm **107** causes piston **124** to reciprocate within cylinder **126**, thereby compressing air as described above in the previous embodiment. When cam **109** is again driven by engine **12**, the bias of locking spring **110** is overcome. Cam **109** again begins to rotate and pushes rocker arm end **105** in the same pivotal oscillation as described above. When longitudinal extension **116** of cam **109** is substantially parallel to rocker arm **107**, latch **106** is pushed away from rocker arm end **105** so that the two are no longer in contact.

It is possible in either of the embodiments of FIGS. **1** and **2** for the cam to be driven by an energy source other than engine **12**, such as an electrically driven motor (not shown). In this case too, spring lock mechanism **104** can be used to hold rocker arm end **105** stationary when the alternative force that drives cam **109** is disabled.

While this invention has been described as having a preferred design, the present invention can be further modified within the spirit and scope of this disclosure. This application is therefore intended to cover any variations, uses, or adaptations of the invention using its general principles. Further, this application is intended to cover such departures from the present disclosure as come within known or customary practice in the art to which this invention pertains and which fall within the limits of the appended claims.

What is claimed is:

1. A compressed air powered motor vehicle, comprising:

an engine including an intake and an exhaust pipe, said engine being configured for operating on compressed air received in said intake and expelling exhaust air through said exhaust pipe;

a compressed air storage container in fluid communication with said intake of said engine;

at least one compression mechanism configured for compressing the exhaust air, each said compression mechanism including:

a substantially hollow cylinder having at least two inlets in fluid communication with said exhaust pipe of said engine, each said inlet including a first check valve allowing passage of the exhaust air into said cylinder while preventing passage of the exhaust air out of said cylinder, said cylinder also having at least two outlets in fluid communication with said compressed air storage container, each said outlet including a second check valve allowing passage of the compressed air out of said cylinder while preventing passage of the compressed air into said cylinder;

a piston slidably disposed within said cylinder, said piston forming a substantially airtight seal with said cylinder, said piston having a first side and a second side, said cylinder and said first side of said piston defining a first chamber, said cylinder and said second side of said piston defining a second chamber, each said chamber being associated with at least one said inlet and at least one said outlet; and

a rod extending axially from said cylinder, said rod having a first end attached to said first side of said piston and a second end disposed outside said cylinder, said rod being slidable relative to said cylinder;

a rocker arm having a first end, a second end and a pivot point disposed between said first end and said second

end, said first end of said rocker arm being attached to said second end of said rod of a first said compression mechanism, said rocker arm being configured for pivotal oscillation about said pivot point, thereby reciprocating said rod and said piston within said cylinder; and

a cam coupled with one of said first end and said second end of said rocker arm, said cam being configured for pivotal oscillation of said rocker arm.

2. The compressed air powered motor vehicle of claim **1**, wherein said piston has a direction of travel, said motor vehicle further comprising a bracket interconnecting said second end of said rocker arm and said cylinder of a first said compression mechanism, said bracket carrying said cylinder and moving said cylinder in a direction opposite to said direction of travel of said piston.

3. The compressed air powered motor vehicle of claim **2**, further comprising a stabilizer bar oriented substantially parallel to said rod, said stabilizer bar slidably carrying said bracket.

4. The compressed air powered motor vehicle of claim **1**, further comprising a fixed structure and at least one suspension spring having two opposite ends, a first said end of each said suspension spring being attached to a respective said end of said rocker arm, a second said end of each said suspension spring being attached to said fixed structure, said at least one suspension spring being configured for limiting said pivotal oscillation of said rocker arm.

5. The compressed air powered motor vehicle of claim **1**, wherein each said chamber is associated with a single respective said inlet and a single respective said outlet.

6. The compressed air powered motor vehicle of claim **1**, wherein said air powered engine includes an output shaft driving said cam.

7. The compressed air powered motor vehicle of claim **1**, wherein said at least one compression mechanism is configured for expelling the compressed air through at least one of said outlets on each of an upstroke and a downstroke.

8. The compressed air powered motor vehicle of claim **1**, wherein said cam is coupled with said second end of said rocker arm, said compressed air powered motor vehicle further comprising a spring lock mechanism configured for holding stationary said second end of said rocker arm when said cam is stationary.

9. The compressed air powered motor vehicle of claim **8**, further comprising a fixed structure, and wherein said spring lock mechanism includes a latch attached to said fixed structure and configured for engaging said second end of said rocker arm, said spring lock mechanism also including a resilient device configured for biasing said latch against said second end of said rocker arm.

10. The compressed air powered motor vehicle of claim **9**, wherein said resilient device comprises a locking spring interconnecting said latch and said second end of said rocker arm.

11. The compressed air powered motor vehicle of claim **9**, wherein said cam has a longitudinal extension, said latch engaging said second end of said rocker arm when said longitudinal extension of said cam is oriented substantially perpendicular to said rocker arm, said cam being configured for biasing said latch away from said second end of said rocker arm when said longitudinal extension of said cam is oriented substantially parallel to said rocker arm.

12. The compressed air powered motor vehicle of claim **1**, wherein said second end of said rocker arm is attached to said second end of said rod of a second said compression mechanism.

13. The compressed air powered motor vehicle of claim **1**, further comprising a third check valve in fluid communica-

tion with said at least two inlets of said cylinder, said third check valve allowing passage of air from an ambient environment into said at least two inlets while preventing passage of air from said at least two inlets into said ambient environment.

14. A compressed air powered motor vehicle, comprising:

an engine including an intake and an exhaust pipe, said engine being configured for operating on compressed air received in said intake and expelling exhaust air through said exhaust pipe;

a compressed air storage container in fluid communication with said intake of said engine;

at least one compression mechanism in fluid communication with said exhaust pipe of said engine and with said compressed air storage container, said at least one compression mechanism being configured for compressing the exhaust air and replenishing said compressed air storage container, each said compression mechanism including a reciprocable connecting rod having a direction of travel;

a rocker arm having a first end, a second end and a pivot point disposed between said first end and said second end, said first end of said rocker arm being attached to said connecting rod, said rocker arm being configured for pivotal oscillation about said pivot point, thereby reciprocating said rod within said compression mechanism;

a bracket interconnecting said second end of said rocker arm and said cylinder, said bracket carrying said cylinder and moving said cylinder in a direction opposite to said direction of travel of said rod; and

a stabilizing bar oriented substantially parallel to said rod, said stabilizing bar being configured for slidably carrying said bracket.

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